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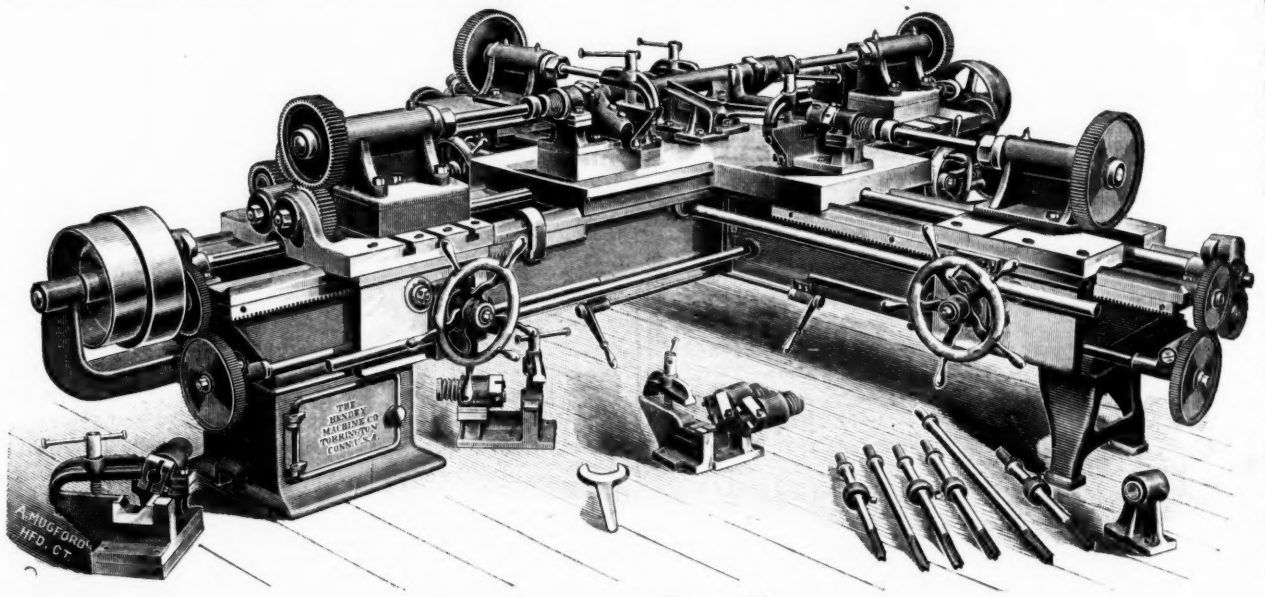
Six Spindle Boring, Drilling and Facing Machine.

The accompanying illustration is of a machine which was originally designed and built for one of our prominent harvesting machinery builders for boring and facing bearings in the frame of a mower, six bearings being finished simultaneously. Instead, however, of making the machine for one frame or set of conditions only it was so designed and built as to adapt it to a very considerable range of work in which holes are to be bored in line with or at right angles to one another, thus adapting it not only to different styles of mower frames, but for other work as well.



BORING, DRILLING AND FACING MACHINE.

As will be perceived, the bed is in the form of a cross, and carries upon each of its four arms a carriage. All these carriages can be moved by hand the same as a lathe carriage, and have also automatic feeds. At the center of the bed or at the intersection of the four arms of the cross, the work is secured upon a platen, to which it is lifted preferably by an air or other hoist, though when not too heavy to be readily handled it can be put in place and removed by hand.

Two of the spindles are adjustable laterally, and all of them can readily be made so if circumstances require it.

The plan adopted in this instance was to arrange for clamping the work uniformly and then place around it doweled brackets which support bushings and facing tools; the latter being driven by clutch collars fastened to the bars.

The principal dimensions of the machine, as shown, are as follows: Extreme length of bed over all, 16 feet 9 inches; length of bed the shorter way, or of the "side beds," 12 feet 9 inches; feed of main heads 25 inches; feed of side heads 18 inches. There are two tables for holding work, one of these being 2 feet x 4 feet 6 inches, the other 1 foot 9 inches x 2 feet. It can be geared to suitable speeds for holes varying from ¼ inch to 2½ inches and to face 4 inches diameter. Its weight is about 9,000 pounds, and it is made by the Hendey Machine Co., Torrington, Conn.

A Street-car Fare Register.

We present on this page illustrations of a new street-car fare register, in the designing of which especial attention has been paid to simplicity and to reducing the number of parts as much as possible consistently with the functions to be performed.

To those familiar with most of the devices of this kind, Fig. 1 will give the impression that behind the large gear shown must be the usual collection of gears, ratchet wheels, pawls, springs, etc., while the fact is there are none of these except the single piece shown in Fig. 2, and the spring wire which lies across it.

The drawing up of this bar when the operator pulls the cord actuates the entire mechanism and produces the following results:

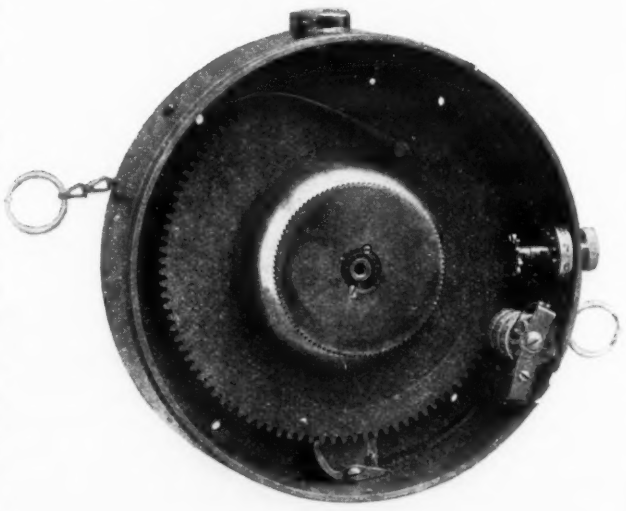


Fig. 1.

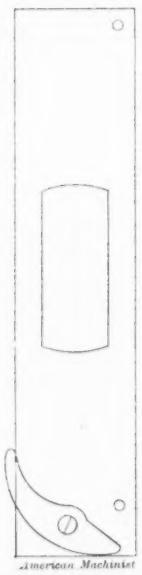


Fig. 2

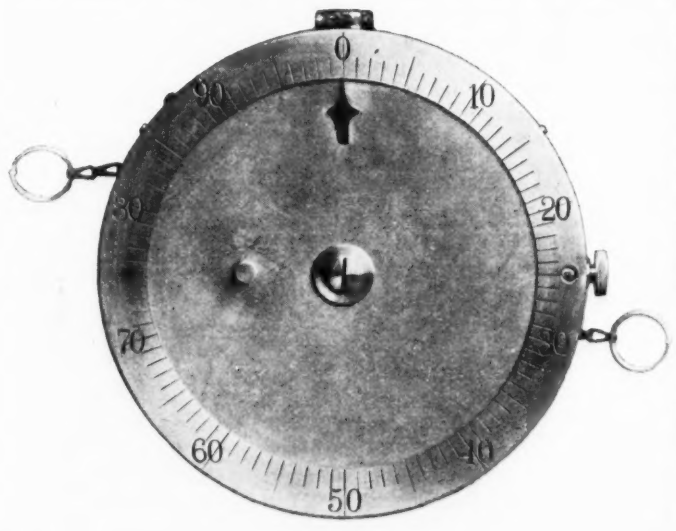


Fig. 3.

A STREET CAR FARE REGISTER.

The pawl shown at the bottom of the bar, Fig. 2, and attached thereto, engages a tooth in the large gear wheel, and causes it to rotate one tooth. In doing this the bell hammer is made to rise, and is then released, and the spring of the hammer arm causes it to strike the bell or gong.

The large gear, which has 100 teeth, is geared with a smaller one which has 10 teeth, and which lies behind the small cylinder shown in the lower right hand part of Fig. 1.

This small gear is always in mesh with the large one, and it is evident that any motion imparted to the large gear must be absolutely transferred to the smaller—so when the large gear is turned one tooth the smaller also rotates one tooth, and when the large gear has turned one revolution the small one will have turned ten revolutions.

The small gear carries a disk on the periphery of which are the numerals 1 to 10, and this disk imparts motion at each revolution to a record disk, which, in turn, advances a third at each turn, which results in recording, consecutively and continually, the number of times the bell is rung, or the large gear is rotated one tooth.

This register or record cannot be changed without destroying the machine or taking it apart. This register is read by the starter while standing in the doorway and looking at the edge of the register.

The face of the machine as seen in Fig. 3 shows that 100 fares can be tallied on it. If more than 100 fares are collected a small dial shown in Fig. 1 to the right will indicate how many hundreds have been passed.

It is, of course, more convenient to return the pointer to 0 when a new trip is started, and, to do this, the small knob, shown near the center in Fig. 3, is pressed in, and the dial is then free to rotate and the pointer carried by it is brought back to 0.

But it must be remembered that this does not affect the consecutive register, and if, while the dial is free, the bell is rung for a fare, the consecutive register will tally.

There are many interesting points about this very simple register, which those who are interested in street cars will find worth investigation. It is an invention which has been developed and perfected by Wm. D. Forbes, M. E., of 1300 Hudson street, Hoboken, N. J., and is manufactured by The Columbia Cash Register Co., at the same address.

Every once in a while some one gets waked up to the overcrowding of street cars in our large cities, and dire threats are made of what will be done in the way of preventing it. Then after a little there is quiet and the overcrowding goes on as usual. In fact, it never stops. The street car companies make money by the overcrowding process and do not care any more for the howling of the people than they do for a west wind. Men and women after a hard day's work clinging to straps for miles are not getting what they pay for, but it is the way to run a railway to make money, and it goes that all the talking in the world will not remedy matters even just a little. Nothing short of a rigid law, as in European countries, defining the numbers of passengers that may be carried in a car, will work any change. And, after all, the people themselves would very likely be the first to complain at such a law. One has only to observe the rush that will be made to overcrowd a car on a line when another car will start in a couple of minutes, to believe this. The proverbial hurry of Americans is at the bottom of the whole matter. The railway companies are simply taking advantage of a national trait. This matter of hurry crops out in a choice of means of conveyance. Given moderate

speed and comparative safety as against high speed and danger and the former isn't in it as it were.

Calculating Weights for Estimates—III.

By J. Horner.

I propose now to take castings of various and of good typical forms, and without going through the actual calculations, which would only be a repetition of what I have already done, show how they may be suitably divided up and averaged for purposes of calculation, when but a fair approximation to absolute accuracy is required.

Fig. 7 shows a friction brake wheel. The size is immaterial in the present connection, the matter for consideration being the method of dividing it up for purposes of calculation. The rim A is taken as a ring unrolled into a rectangular strip of metal, equal in length to the circumference, of width a and thickness b. The sides of the brake wheel comprise outside flanges, internal rims, arms and boss. These need not be divided. Instead of taking annular rings of width c, and arms of width d, I should take a disk of diameter B, and simply deduct the openings e from it. But I should not reckon these openings as triangular, because that would involve some little calculation. I should strike a circle, e e, having an area very approximately equal to that of the triangular opening (the eye alone judging of the amount of approximation required), and deduct the area of four such circles, obtained from a table, from the area of the disk, of diameter B. There would, of course, be two such disks in the brake wheel. In the matter of bosses it is by no means usual in approximate calculations to take the mean diameter of the annulus of metal in the boss, and thence deduce the circumference. It is easier and usually sufficiently accurate to take the boss as a solid lump and omit the cored hole in the calculation. The amount of metal taken out by the hole does not amount to much, and usually something else, as the small hollows or fillets in the pattern, can be set off against it. So that taking the boss as solid without a cored hole, nothing then need be added for the fillets. For the bosses then I should take a solid disk of diameter f, and of thickness equal to the added thicknesses of the four thin bosses, two outside the plates and two inside. The ribs g and h are merely plain strips of definite length, width and thickness, and are so taken.

Fig. 8 is a bevel pinion, cast with a friction cone, as used in the hoisting gear of some cranes. I should divide this out as indicated by the shaded sections A, B, C, D, judging by the eye of the averaging of metal. A includes the cone, B takes in the narrow ring of metal which unites the cone to the wheel, and is brought for convenience into a ring of rectangular section. C averages a portion of the teeth in the annulus of metal round the shaft hole, taking due account of the clearances or teeth spaces, D averages the rest of the teeth, also taking account of the teeth spaces.

The weight of a bracket like Fig. 9 might be obtained in the following way: If it were necessary to obtain the weight exactly the plate A, of thickness a, would have to be divided into suitable triangles and one rectangle, as indicated by the dotted lines, and figuring 1, 2, 3, 4, 5. This would take some time, so for most purposes, a rectangle might be averaged of mean length b and mean width c. Strictly, this area of the plate should be reckoned as inside the foot B and the ribs C and D, but I should take the outside dimensions and still take the foot and rib at their full width, that is without making any deduction for the thickness a of the plate. This will make an allowance for the fillets.

The foot B is a plain rectangle of length d, and width e, and thickness f. The rib C is taken of length g, of mean width h, and of thickness i. The rib D is taken of length j, of mean width k, and of thickness l, making no deduction in either case for the thickness a of the plate A.

The bearing E may be safely taken as a ring of metal, outlined in elevation, of thickness m, the slight excess of metal compensating for sundry odd corners and angles not taken account of otherwise.

A crank like Fig. 10 can be averaged as follows:

Two bosses, A and B, of diameter a, and depth b, which may be considered either as solid disks, or as rings. The web C of mean length c, and mean width d, and thickness e, and a rib D, of length f, and mean width g, and thickness h, completing the crank. A crank disk like Fig. 11 can be conveniently divided as follows:

The plated portion A can be taken as a plain disk, of diameter a, and thickness b. The flange, or ring B, is taken as a ring of

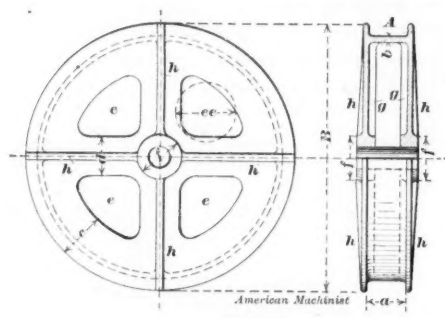


Fig. 7

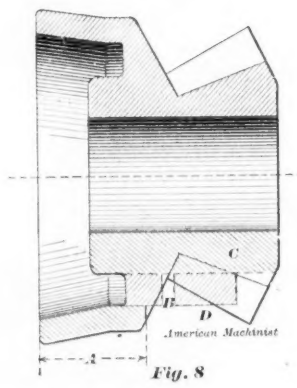


Fig. 8

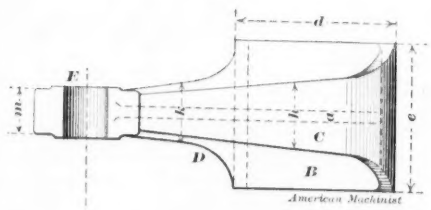


Fig. 9

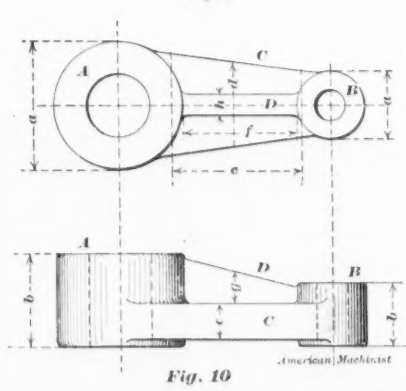


Fig. 10

cast round the internal angles of the disk, which would be about correct.

In the double bracket, Fig. 12, there is the base A, of length a, width b, and thickness c. The vertical plates B B may be taken as triangular pieces, as indicated by the prolongation of the slant edges, and deduction made on account of the portion of the triangle shown prolonged. But I should take the mean width d, and so form a rectangle of width d, and length e, and thickness f, and so simplify the calculation. The ribs C would be taken of length g, and width h, and thickness i. The bosses D would be reckoned as solid disks, of thickness j, minus thickness f, and the holes would be set off against fillets in the casting.

Figs. 13 and 14 show a center cross, which was used for a special type of revolving crane, upon which the whole superstructure of the crane swiveled. I will show how the casting is conveniently divided for getting at the weights. Fig. 13 is a half plan view of the casting, and Fig. 14 comprises on the

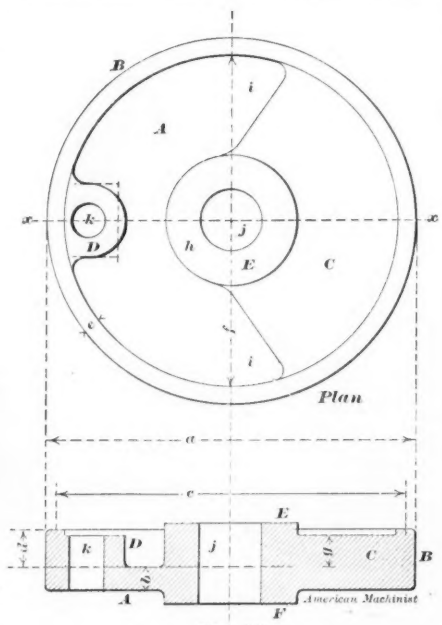


Fig. 11

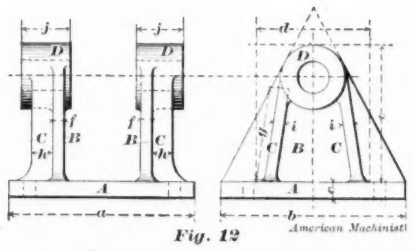


Fig. 12

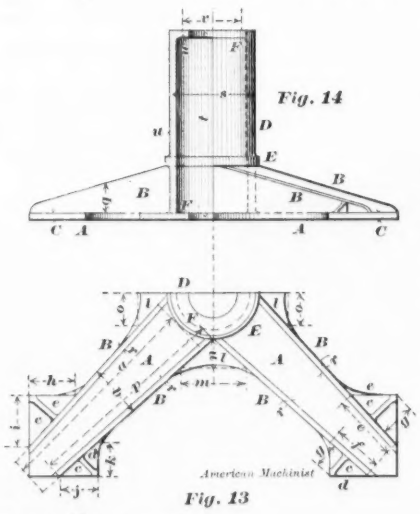


Fig. 13

Fig. 14

take half the product. And similarly the base e of the triangle e, into its height g. As there are 4 feet, averaging the triangles into half the number of rectangles (left-hand side of figure) h and i would form two sides of a rectangle. Similarly j and k would form two sides of a smaller rectangle. To obtain the areas of these, the length and breadth of the two sides would have to be multiplied together. So too the curved portions l, l would be treated either as triangles of length m, and height n, or else as rectangles, having sides equal in length to o.

The eight vertical ribs B are measured of length p, of mean width q, and of thickness r. There are eight triangular brackets C, abutting against these ribs, and these may be measured as triangles, or as rectangles. The body of the center pin boss D is taken as a cylinder of mean diameter S, of length t, and of thickness u. There is a shoulder or ring E, similarly averaged. There is also an internal annulus or ring of metal F, at each end of the pin, of mean diameter v, and thickness w, and these complete the dividing of the casting.

Fig. 15 (page 5) illustrates a center casting for a large revolving crane. It pivots around the center pin of the square base last illustrated, and carries the main girders upon which the entire superstructure of the crane is borne. The views comprise a plan, and end view, and a half elevation, and part section, the latter along the line x—x. The dimensions are conveniently averaged as follows:

The main plate or web A can be averaged as a rectangle, of length a, width b, and thickness c. The top plate can be averaged into one plain rectangle, or as follows:

Take the area of a circle B, of diameter d; two rectangles C C, of length e, and width f, and two other rectangles D, D, of length g, and width h, which is averaged to include the portions i and j. B, C, D are each of the same thickness, K.

The bottom plate may be divided up into two portions. First, E is a circle of diameter l, equal to d above, but the area of the central hole, of diameter m, must be deducted from it. Then the two portions F, F are each taken, of length n, and width o. The radii p and q can be averaged as triangles or rectangles. The thicknesses of E, F, p, q, are equal to r. There are two ribs G, G. These are plain rectangles, of length S, width t, and thickness u. The brackets H, H are taken as two triangles, or as one rectangle. Two brackets J, J against the main boss are taken of length v, width w, and thickness x. The small internal boss and ribs K are taken, the first as an annulus, the second as rectangles. The main boss L is taken as a cylinder of mean diameter y, and length z, and thickness a a.

There is a very great deal of rough averaging done in the estimation of weights. Measurements of areas and of volumes have to be taken with great expedition, because the element of time presses so heavily. As a rule, all rules of mensuration which involve much calculation are passed by, and figures are averaged into rectangles, triangles and circles, or parts of circles. For the areas in the first case two dimensions only have to be multiplied together. In the second the base and height, and half the product taken. In the third, a table would be consulted. The simplification of irregular figures is done by means of a scale, or a rule applied to the drawings, and in many cases the eye judges of the approximation done. In some of this averaging of parts, it is an assistance when the drawings are made to scale, to cut out pieces of paper of the shape of one portion, and turn them over upon another portion, which it is desired to average, and note, and allow for any discrepancy that may exist. Some time may be saved in this way, and much practical accuracy secured. Rough pencil tracings may be made for the same purpose.

Calculating Weights for Estimates.

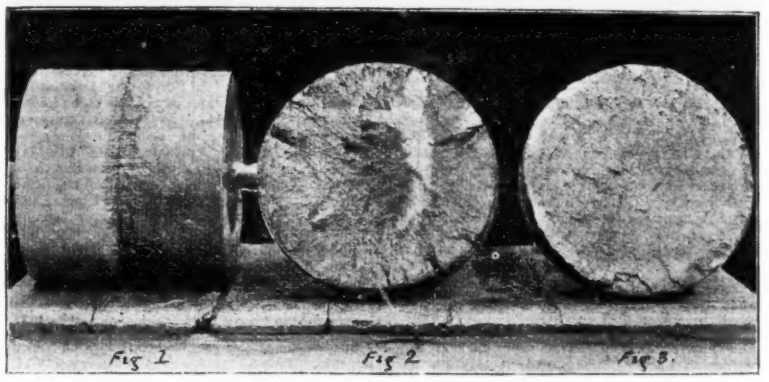
Mr. John McGavin, of Binghamton, N. Y., claims to have invented a method of propelling vessels by compressed air, and expects to accomplish great things by its use. We have not yet heard of the result of a trial of it.

The Corrosion of Tail Shafts in Steamers.

One of the most common mishaps at sea is the breaking of a shaft, and remarkable ingenuity is often shown by the engineers in making some sort of repair sufficient to enable the boat to reach her destination. When the tail end shaft breaks the matter is particularly serious, as repairs are then usually out of question. The main cause of fracture at this spot is undoubtedly galvanic action between the brass liner and the iron shaft, and the annexed engraving, reproduced from a photograph, shows this very clearly. In Fig. 1 a somewhat early stage in the process of corrosion is exhibited. The piece of shafting shown was covered with a brass liner on the left, the right hand portion being exposed to sea water. After being used for two years only, it was found in the state exhibited, and had to be removed. The peculiar feature is the way in which the corrosion is localized, there being but little rust on the general body of the shaft between the liners. The most striking feature, and the one which makes this kind of corrosion especially dangerous, is the deep narrow groove, like a knife-cut almost, which has been eaten away round the shaft, substantially reducing its effective diameter at this point. Another instance of this is shown in Fig. 3, which has been taken from a condemned shaft which was broken for further information. The groove in this case is about 1 inch deep, and extends half-way round the shaft. The effect of a groove of this kind in weakening a shaft is known to be very serious, particularly in ordinary working. In the testing machine the reduction of strength is less marked, owing to the plasticity of the metal, but when the shaft is doing its daily work, all shock and bending action is concentrated on the reduced section, fatiguing the metal, with the result that the shaft may go to pieces finally under quite a low nominal load. This is very clearly shown in Fig. 2, which represents a section of a shaft removed after six years' service, and broken by dropping a ball on it. Externally, this shaft showed no worse signs of deterioration than Fig. 1, but the corrosion was more general over the body of the shaft, as it had run longer, and this greater extent of rust somewhat masked the damage done on the section just behind the liner. On fracture, however, the only sound metal was the small light-colored vertical strip visible in the engraving; the whole of the rest of the section was fractured and discolored. It was remarkable that in these conditions it was possible to lift the shaft out of the boat without it dropping to pieces, and yet it had been driving the propeller, and was not suspected of being anything like so bad. Of course, the whole of this deterioration was not directly due to galvanic action, which was responsible in the first case only for cutting a narrow groove round the shaft. The rest of the damage was then done by the concentration of the stresses due to shocks, bending, etc., on this reduced section, with the well-known results of gradual deterioration of the metal. This local corrosion is not uncommon in iron when the latter forms part of a galvanic circuit. A familiar instance is the pitting of boilers. These pits are found to be filled with iron peroxide, which is electro positive to iron, and unless the last fragments of this peroxide are cleaned away from the pit, the corrosion will be renewed, but, as is well known, is localized in the pits, and does not extend over the body of the iron.

We do not know if an attempt has yet been made to protect these tail shafts from this galvanic action by means of zinc, but, if not, the experiment might be worth trying.

Another device for the same purpose has been brought forward by Mr. Mudd, of the Central Engine Works, Hartlepool, who fits an elastic cover over the tail shaft, to which it is cemented. This cover overlaps the liners at each end by five or six inches, and completely excludes the water from the shaft. Under these conditions, corrosion, whether galvanic or otherwise, should be effectually prevented, and the life of such shafts materially prolonged.—*Engineering*, London.



Corrosion of Tail Shafts.

Universal Milling Machine.

We illustrate on this page a universal milling machine made by the Lodge & Davis Machine Tool Co., Cincinnati, Ohio.

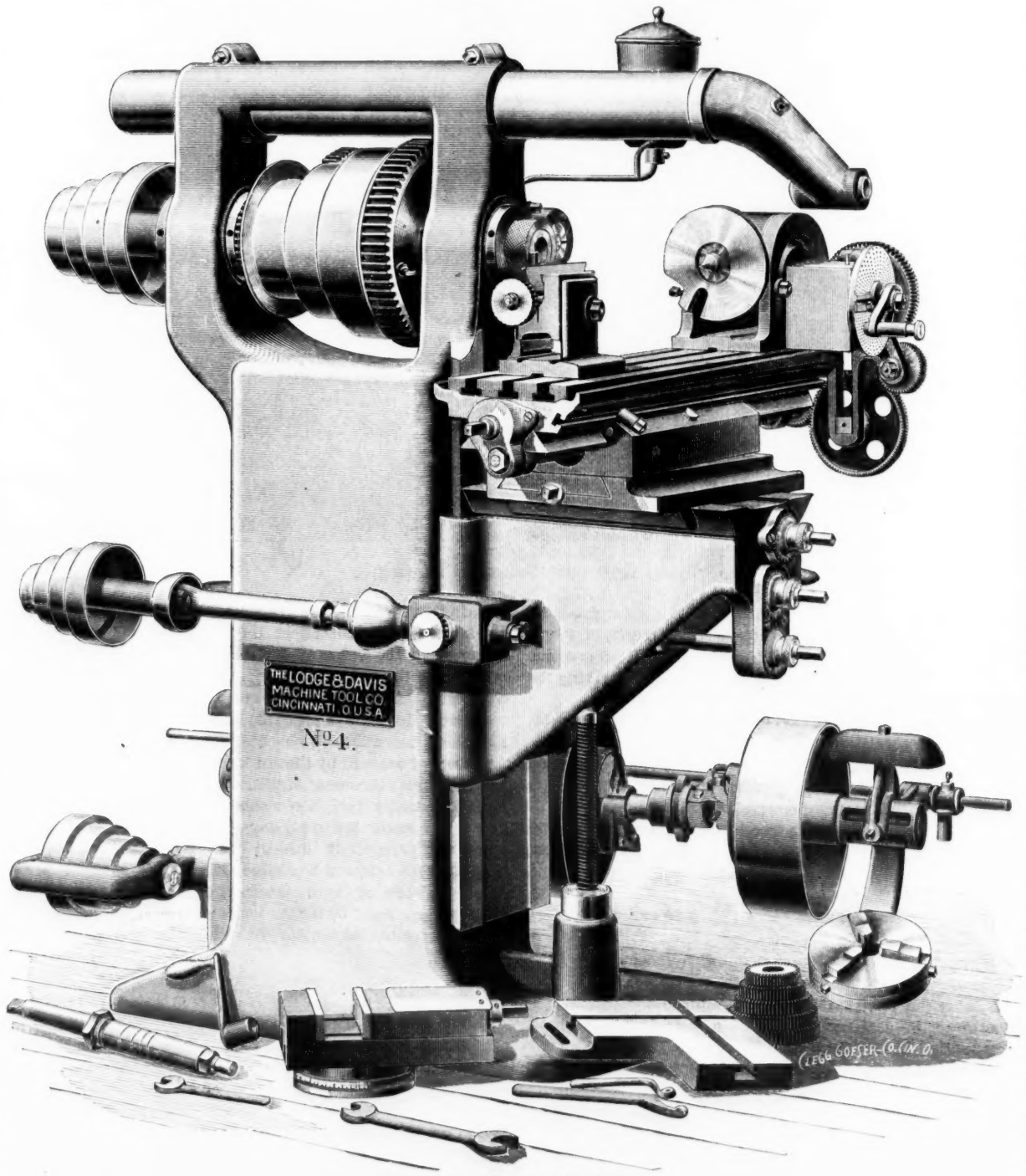
An adjustable overhanging arm is provided and fits in clamp bearings over each spindle bearing, the uprights being independently tied together, as shown. The spindle is of crucible steel, is hollow and runs in phosphor-bronze boxes adjustable for wear. The end of the spindle is threaded to take the same chuck that fits also the spiral head spindle.

The table has three T-slots running its entire length, and around it an oil channel is formed which catches all oil and conducts it to a drainage tube and suitable vessel which can be attached to it.

The foot stock center is adjustable vertically for taper work, and the spiral head has the usual adjustments above and below the center line. It is made as compact as possible and is provided with plates for indexing up to 360 teeth. Gears are provided for cutting spirals from 1.5 inches to 66 inches pitch.

The table is fed automatically in either direction at 12 different rates relative to spindle revolution, ranging from .007 inch to 1 inch, and the feed is reversed by turning the knob seen at the left of the knee. It is automatically stopped by the stud, which is adjustable in the T slot, and operates in either direction. The lower feed cone is upon a swiveling yoke, which permits it to be adjusted to secure proper tension on both feed belts. The table can also be moved by hand either from the left hand end or from the front of the knee, and it has a quick return motion. The handle by which the feed is thrown, and by which it may also be thrown out if desired, is at the right of the knee.

All traversing screws are provided with micrometer graduations, and the dial plates are adjustable, and are held in place by knurled nuts, which are readily loosened or tightened by the fingers. Taper gibs are provided for all sliding surfaces. The machine is provided with all the parts and attachments shown in the engraving.



Universal Milling Machine.

The Efficiency of Centrifugal Pumps.

The results of some experiments on the efficiency of centrifugal pumps recently made on the Continent have been published, and are of interest as indicating how frequently the conclusions arrived at theoretically require modifying in practice. The design of the pump was such that the discharge through the wheel casing was taken through a narrow orifice extending around the entire circumference of the casing. This orifice gradually widened outward to allow the water to arrive without shock in a spiral collector surrounding the pump and leading into the discharge pipe. The width of the orifice was such as to give the required flow at the velocity due to the head against the pump, with an allowance of 0.6 for contraction. The maximum mechanical efficiency of the pump was found to be only 58 per cent., and occurred only with the greatest flow—about 54 gallons. The maximum efficiency for any given rate of delivery occurred when the lift was between 50 and 60 feet. Another pump was constructed after these trials, in which the form of the blades and of the casing was altered from that first adopted, and with this pump an efficiency of 65 per cent. was obtained with a discharge of 6.6 gallons under a head of 65.6 feet, and an efficiency of 68 per cent., with a discharge of 7 8 gallons under a head of 32 8 feet.—*The Mechanical World.*

F. L. O. Wadsworth, the author of the paper on "Manufacture of Accurate Straight-Edges," whose address we gave in our issue of September 6th, as The International

Bureau of Weights and Measures, writes us that his address is now The University of Chicago, Chicago, Ill. This announcement we make for the convenience of those who may wish to communicate with Mr. Wadsworth in relation to the pamphlet which we mentioned.

Machine Shop Milling Practice—XIII.

By Horace L. Arnold.

In the month of February, 1894, the Brown & Sharpe Co. instituted a series of experiments intended to determine the question, whether the mill should run against the feed of the work, old style, or with the feed of the work, new style, or "*on*" or "*off*" the work, as the two methods are designated in the B. & S. shops; "on" meaning the direction of revolution which causes the mill teeth to strike *on* the top surface or "scale" of cast-iron work in process of being milled, or the "new" way; while milling "off" means revolving the bottom of the mill against the feed, so that the cut begins in the clean, newly cut surface of the work and cuts by breaking through the scale on the untouched surface, which is the old style. As has been previously fully discussed in the American Machinist, the "old," or "off" way, commences the chip on nothing and ends the chip at its greatest thickness, while the "new," or "on" way, begins the chip at the greatest chip thickness and ends it on nothing.

The machine selected for this "off" and "on" trial was the No. 5 plain milling machine (old No. 8), the heaviest of the pillar and knee machines made by the B. & S. Co. This machine weighs about 5,500 pounds, stands in 115x69 inches, has a 4½-inch belt on a 13¾-inch pulley, and is backgeared, which gives a very strong spindle drive, spur geared throughout, same as were the Pratt & Whitney machines, on which the "on" feed showed to great advantage over the old "off" method. The table of this machine, shown in Fig. 72, has a longitudinal feed of 48 inches, traverse movement 9¾ inches, and vertical action to 19½ inches below center of spindle. As will be seen from the engraving, the overhead arm is heavy, the out board arm support is substantial, and all the mill arbor support possible with the pillar and knee form of machine is obtained.

This No. 5 machine has given extremely good results where substituted for planers. In the Thorne typesetting machine shop, May, 1894, with 2 mills, each 5 inches long, 4½ inches diameter, 15 teeth, radial face, spiral 2 inches in 12 inches, teeth cut away at intervals, as shown in Fig. 71, issue of August 30th, the following results were obtained in regular work:

Revolutions of cutter, 42.
Surface speed, 50 feet per minute (nearly).
Fastest feed, 9½ inches per minute.
Number teeth per minute, 630.

Feed per tooth, 0.015 = $\frac{1\frac{1}{2}}{100}$ inch.

In annealed cast-iron:

Average feed, 8¾ inches per minute.
Average per tooth feed, 0.014 inch (nearly).

In unannealed cast-iron:

Average feed, 6 inches per minute.
Average feed per tooth, 0.0095 inch.

The anneal was not stated; probably the work was packed either in hematite or in cast iron chips.

In annealed cast-iron stock mills cut (average per tooth chip 0.014 inch greatest thickness, surface speed 50 feet per minute, cut dry) 1,550 inches without grinding. This gives 3 hours run, at the end of which the mills were, seemingly, in same condition as at the end of first hour. This work was finished with 1 cut, some of it, and some with 2 cuts.

The limit of variation for the work was same as limit for same work done by planers; for one cut only, limit of variation from a perfect plane $\frac{1}{1000}$ inch, with 3 points on test plate = actual limit of variation from plane $\frac{1}{1000}$ inch.

Where work was finished with 2 cuts the limit of variation from a plane was $\frac{2}{1000}$ inch, with 3 points on test plate = $\frac{1}{1000}$ actual variation, or, in other words, the depth of grinder cut needed to bring this work flat would be 0 0 inch to 0.001 inch.

The unannealed work was in the form of plates 19 inches long, 7¾ inches wide by ⅝ inch thick. The annealed work was 16 inches long, 9½ inches wide by 1¼ inches thick. Work was *all* within the limit; much of it was nearly perfect, and the limit of imperfection was smaller with the milling machine than with the planer, and the planing at the Thorne shops is of a very superior order.

It will be noted that these plates, 7¾x19 inches, and 9½x16 inches, were surfaced in about *two minutes* each. This amazing speed will give those who are accustomed to the use of the planer only some idea of the advantages of milling under favorable conditions, and will prepare the reader for the statement made by the Thorne Co., that the milling machine could do as much work in 2½ minutes as the planer could do in 40 minutes, or that the *time* cost was as 16 to 1 in favor of milling over planing.

From the foregoing it will be seen that the Brown & Sharpe Co. had the best of grounds for their belief that the No. 5 had ample strength and stiffness to give a decisive test of the respective merits of "on" and "off" milling. With the No. 5 machine the following tools were used in making the test:

1 mill arbor, 2 inches diameter, with collars 2½ inches diameter.

Several mill arbors (same being destroyed in the test) 1½ inches diameter, with collars 2 inches diameter.

These arbors all had the same shank in driving spindle, taper ½ inch to the foot, 1½ inches diameter at large end, and 6 inches long.

2 cutters (or mills) 6 inches face by 4½ inches diameter.

2 cutters (or mills) 6 inches face by 3 inches diameter.

Number teeth in mills, 15.

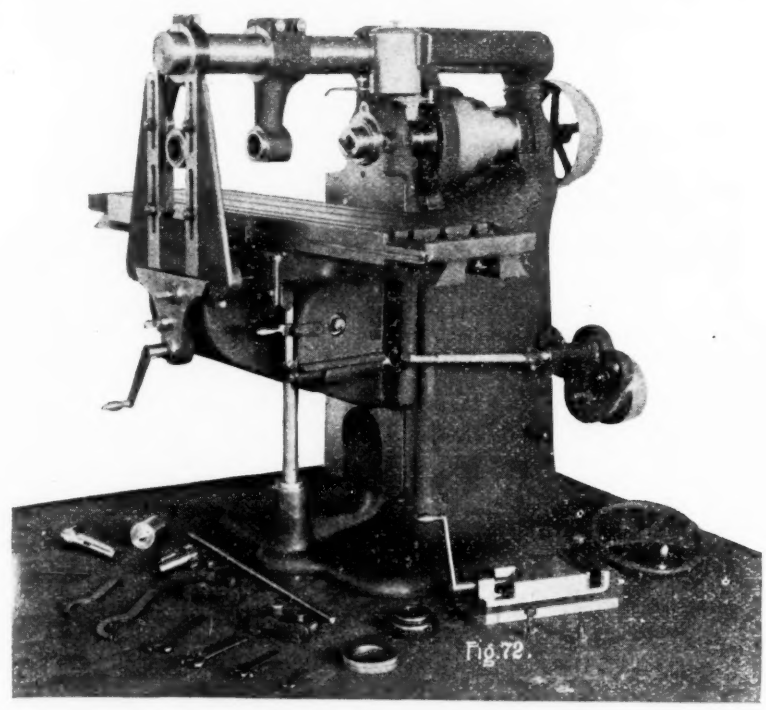
Number revolutions per minute, 42 in all test cuts.

Surface speed of 4½-inch mills, 50 feet per minute (nearly).

Surface speed of small mills, 33 feet per minute (nearly).

The stock on which the mills were tested was unannealed cast iron, pickled down to perfectly clean iron, every trace of sand removed, in bars about 36 inches long by 6 inches square, cast with a core 3 inches square through the whole length. These figures for dimensions of stock bars are given from memory, and may not be exact. The hollow square section made the bars sound and easy to clamp down on the table.

All of the cutters were splined on their arbors, and all the test cuts were made with the supporting arm 12 inches from the spindle nose, thus giving all possible support to the mills. The test cuts were made as follows: There were two cutters, say "A" and "B," of each diameter; a test bar of pickled unannealed cast-iron being clamped to the milling machine table, the cutter "A" was used to make one cut the length of the test bar, 36 inches, the teeth of the cutter running "*on*" to the work or *with* the feed; then the test bar was turned a quarter way, clamped, and the next side cut with the other mill, "B," running "on" the work, or with the feed of work; next the condition of the mills was carefully noted, and both mills were taken to the grinding machine and made perfectly sharp; then the bar was again turned one quarter way, clamped down and mill "A" used to surface this third side with the mill running "off" the work against the feed, the old way, so that the scale was *broken* from below, not *cut* from above; finally the bar was once more turned and clamped and cutter "B" was used to make an "off" old style cut against the feed. Thus each test bar had two sides finished with each mill A and B, and each mill cut one side running with the feed of the table, and one side running against the feed of the table, thus making it certain that no accidental variation of either stock or tools could affect the inferences drawn from the action of the cutter in any one cut, or the condition of a cutter at the conclusion of surfacing any one side of a test piece. In point of fact, this test showed that this precaution of checking the performance of one mill by that of its mate was a superfluous refinement of distrust, as both mills constantly told the same story. In every cut made the new way the cutter was very perceptibly dulled; the edges of the teeth showed a distinct line, and gave evidence of hard service generally at the end of a cut 30 inches long; so great was the seeming advantage of milling under, "off," or against the feed, old way, that 24 30 inch cuts milling "off" did not seem to injure the cutter nearly so much as *one single cut* milled "on" the new way. Every care was given to this test; the cuts were all ¼ inch deep except in one case where the machine was forced to the limit of its driving power with a quick feed, and stopped with a depth of cut of ¼ inch feeding "on," the new way, as against 1/32 inch depth of cut, same everything, feeding the *old* way, this test showing an effectiveness of 11 with old way against 8 with the new way of feeding. In a test for the endurance of the arbors the machine was not able to distort either arbor when feeding "off" the old way, but was able to and *did* twist and ruin the 1½-inch arbor when feeding "on," or the new way. The quality of the work was about the same with feed in either direction.



Machine Shop Milling Practice.

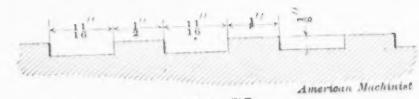


Fig. 73

The revolution marks were 6 to the inch, which, at 42 revolutions per minute, would give a feed in these tests of 7 inches per minute with mills 4½ inches diameter, 15 teeth; the surface speed was 50 feet per minute, and the cut per tooth $\frac{1\frac{1}{2}}{1000}$ inch, or just over $\frac{1}{100}$ inch.

From this test, which certainly appears in every way thoroughly practical and fair, the Brown & Sharpe Co. have decided, as the conductor of this test put it in words, that it is better "*with the Brown & Sharpe machines*" to mill the old way against the feed. Why this test should have shown results precisely and beyond reconciliation opposed to the conclusions drawn by the Pratt & Whitney experts from constant use of milling machine feeds in *both* directions, is more than I can tell. On the face of affairs it would seem that any decision sustained by the practice of either firm of milling machine makers should be accepted as established. It does not seem possible that the pillar and knee form of machine affords a so-greatly different support to the mill at work from that given by the bed and head and tailstock plan of construction, as to make the results obtained from one machine contradict the results obtained from the other. The contradiction is present, however, and I know of no tribunal except that final one of survival through a long period of use which will conclusively decide the question.

The teeth of the mills with which this test was made were notched, as shown in Fig. 71, before referred to. This tooth is cut as follows: First, the teeth are cut with radial faces, spiral, inclined 2 inches in 12 inches; next the mill, 4½ inches diameter with 15 teeth, was put on an arbor, and the notches were cut in the teeth ⅛ of an inch deep with a tool 1/16 inch wide, in the lathe, in a triple spiral 6-inch pitch.

Compressed Air for Cleaning Carpets.

By Frank Richards.

Among the many successful and established applications of compressed air it is a rather striking fact that the success and value of so many of them are each dependent upon some different property of the air or upon some peculiar facility of applying it to the special purpose possessed by no other means. I have lately cited elsewhere in the American Machinist the reliability and the constant and unfailing readiness for duty that leads to the adoption of compressed air for operating switches and signals and crossing gates upon our railroads, also for train brakes, for automatic sprinklers for extinguishing incipient fires in large buildings, for general hoisting and all kinds of intermittent service in extensive and enterprising industrial establishments.

The application of compressed air to the cleaning of carpets is a line of service widely different from that cited above, and, of course, is dependent for its success upon other and different properties. The peculiarity of compressed air which adapts it so completely to the special duty of cleaning carpets we may characterize as its ability "to get there." Everybody knows all about, and too much about, the old way of "cleaning" carpets. Although it has prevailed so long, and prevailed so universally, it is still, when we come to think of it, a curious illustration of how not to do a thing. The old system attacks the carpet instead of attacking the dirt. And how the poor carpets have suffered. When the carpets are cleaned by hand, how unmercifully they are beaten and shaken and abused and mutilated, and when the work is done, how does the exacting housewife still and rightfully complain that the carpets are not clean. And then in establishments that make a business of cleaning carpets we find the same antiquated practice still prevalent. We find them

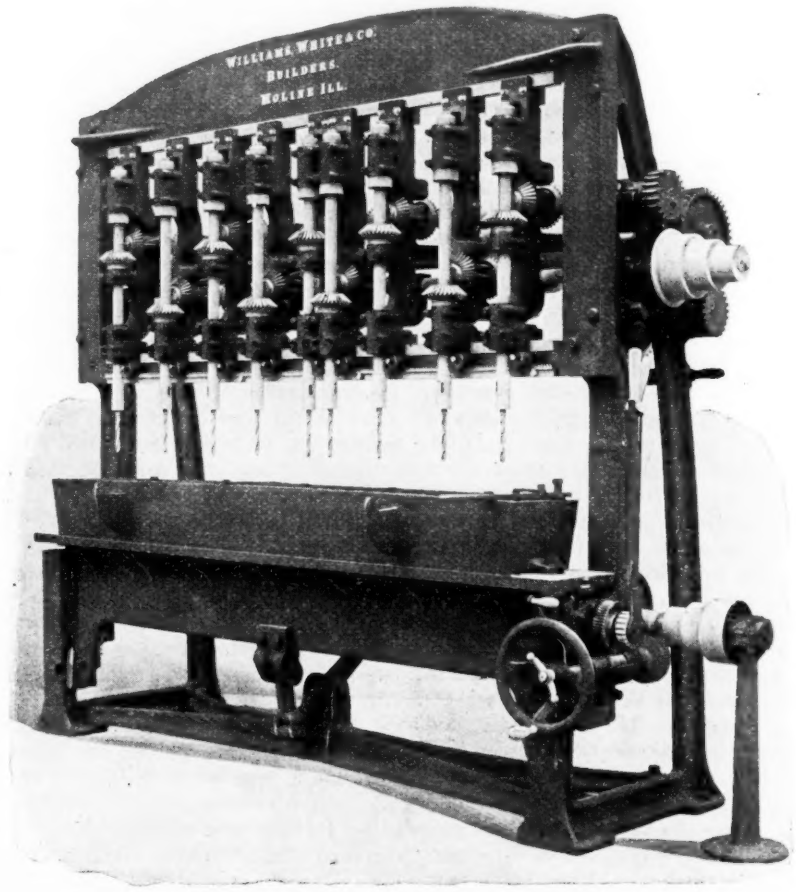
equipped with elaborate systems of machinery, all designed upon the general principle of abusing the carpet as much as possible. Some have series of whips or beaters that successively attack the whole surface of the carpet as it is passed under them by systems of rollers and aprons. In some establishments they use enormous drums or tumblers, 10 or 12 feet in diameter, into which the carpet is thrown bodily and tossed about as long as may be considered safe without too serious injury to it. In connection with the carpet torturing apparatus, the mild and gentle zephyr from an exhaust fan is usually employed to carry away the dust. Under the old system, even the exhaust fan does not get a chance at the carpet itself, but only takes the dirt after it may have been liberated by other devices.

Quite recently I called upon a lady at her house and I found no carpet in her front room. She told me she had sent it to the carpet cleaners, and they had reported to her that the carpet had been "accidentally" injured, and so badly that they were not willing to allow her even to see it, but authorized her to go to the store and pick out a new carpet, which the carpet beating company would pay for. Here, evidently, in adjusting the compromise between carpet and dirt, the carpet got the worst of it.

Now, when you put compressed air at this work, its ability "to get there" enables it to "go for" the dirt instead of for the carpet. When the old system has done its best—by doing its worst—upon the carpet, and confesses its inability to get any more dirt out of it, compressed air will still find dirt in it, and get that dirt out of it, and only then will the carpet be clean. Compressed air for this purpose is clearly common sense. The air does the work not by any *hocus pocus*, but by the actual application of power just where it is required. The power is applied directly to the dirt and it has to "git up and git." The air as most successfully applied to carpet cleaning is used at a considerable pressure, usually from 60 to 100 pounds by the gauge, and when we remember, as I have more than once mentioned in the AMERICAN MACHINIST, that a $\frac{1}{16}$-inch hole freely discharging from 80 pounds pressure has 1 horse power behind it, we have the kick of the horse pretty well concentrated, and when this pungent jet of air permeates the fibers of the carpet it carries great power with it, and the power is actually and directly applied to the expulsion of the dirt. In beating a carpet we may easily believe that some of the more minute particles of foreign matter which it is most desirable to get rid of may be driven into the finer interstices of the carpet instead of being driven out of it. In the use of compressed air, when it picks up any particle, that particle must go where the air goes, which will be through the carpet or out of it in some way. In the case of carpets cleaned by compressed air, a freshness and newness of appearance is observed that does not result from any amount of shaking, or tumbling, or beating. There is an explanation for this. The air employed is preferably used directly as it comes from the compressor, with all the heat of compression remaining in it, so that there is little loss of effective volume from the cooling of the air, and the air in the jet as it first leaves the pipe and strikes the carpet is still more or less hot, and the heat of it is very effective in softening and removing spots of grease or other matter affected by heat, and this action of the air undoubtedly to some extent accounts for the freshness of appearance which the air imparts to the carpet, and is an additional evidence of its penetrating and cleansing power. It is not necessary to point out the value of this process from a sanitary standpoint The thorough and perfect aeration of every particle of the fabric, and the complete removal of all traces of filth and all germs of disease and foulness, would seem to be all that the most advanced and exacting demands of science could require.

The use of compressed air for cleaning carpets has been carried far beyond the theoretical or experimental stage, and is a pronounced success, and an established and permanent business rapidly developing and everywhere prospering. In London, and in all the large cities of Great Britain, establishments employing this process have been opened, and now the carpets of the wealthy classes, especially of the nobility and gentry, and of royalty, are cleaned by compressed air, and can never more be cleaned by any other means. In the United States the development of the process has been along a somewhat different line, but the success of it has been not less pronounced, and the general adoption of it promises to be far more rapid.

The Union Pacific Railroad, and some other railroads in the West, commenced using compressed air for cleaning car seats and found the process so much quicker, cheaper, more effective, and in every way more satisfactory than the old process that its use for this purpose is being rapidly and generally adopted. The practice, as developing here, promises to accomplish better results by simpler apparatus than has been employed in England. Not only are carpets, rugs, cushions and similar articles cleaned by compressed air, but it seems to be generally applicable to the removal of dust and all loose foreign matter from all textile materials, and it would not be a strange thing if we soon found it in the second hand clothing business.



MULTIPLE SPINDLE DRILLING MACHINE.

Multiple Spindle Drilling Machine.

We illustrate herewith a multiple spindle drilling machine which is made with any desired number of spindles, more or less than shown, the particular machine which forms the subject of the illustration, having nine spindles, and being designed for medium work, is capable of drilling nine $\frac{3}{4}$ inch holes in steel, simultaneously, or four 1-inch holes.

The spindles are adjustable vertically to compensate for different lengths of drills. And horizontally they are adjustable to within $3\frac{3}{4}$ inches of each other.

By loosening the clamping screws, the spindles in their saddles can be moved along easily by the pinion and rack, and this can be done without stopping the machine, if desired. The top box carries the weight of the spindle, and a cap at the top takes up all end play, bronze washers on either end of the box reducing the wear.

The feed is obtained from the table, which has 12 inches of movement up and down. There is power feed and automatic stop, quick hand feed by the wheel, and very quick hand feed by the ratchet lever. The automatic feed can be set to throw out the feed when the drills have passed through the work, whereupon the table will drop down of itself, leaving the machine ready for the next piece, and work can thus be very rapidly taken out and put into the machine.

The box which is shown bolted to the table is for drilling under water, but can be removed where longer work must be drilled than table will take in. Inside the box is a hand clamp, by which when a bar or pipe is put in, the whole length can be clamped by a turn of the lever.

The manufacturers are Williams, White & Co., Moline, Ill.

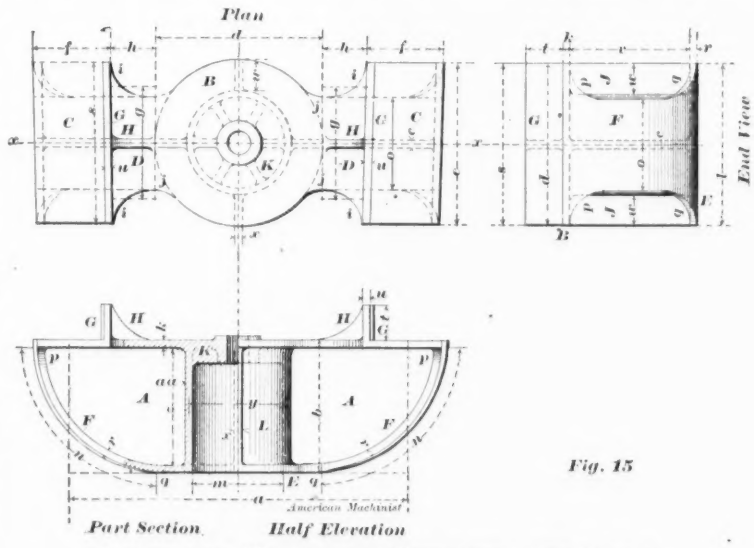


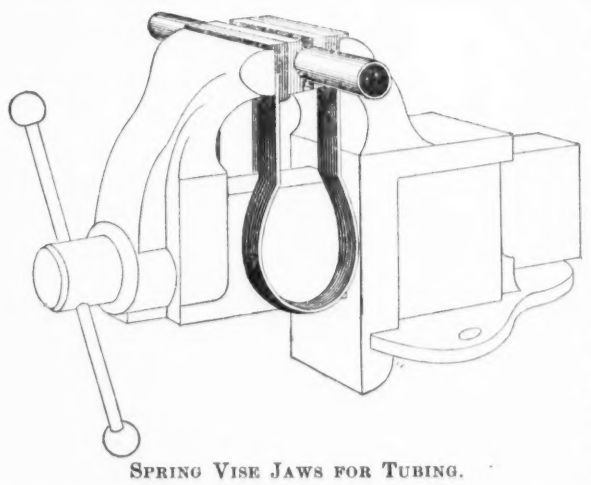
Fig. 15

CALCULATING WEIGHTS FOR ESTIMATES.—SEE PAGE 2.

Spring Vise Jaws for Tubing.

We illustrate herewith a device, the convenience and usefulness of which will be readily understood by mechanics. It is for holding tubing or any cylindrical piece which is too delicate or too highly finished to be held in a pipe vise. The jaws are made of the best vulcanized fiber, securely fastened to pieces of malleable iron, which have shoulders to catch on the jaws of the vise, and they are held together by a spring connection. When the vise is loosened to take out or shift the work, the jaws open out, following the motion of the vise. The paper jaws nearly encircle the tubing holding it so firmly that it cannot slip, and with no danger of distorting the tube, or marring the finish of the surface.

The grooves in the jaws are made to suit any required diameter up to and including $1\frac{1}{2}$ inches. The manufacturers are The Newark Machine Tool Works, Newark, N J.



SPRING VISE JAWS FOR TUBING.

Some time ago it was charged and denied that the U. S. cruiser "New York," owing to depth in the water in excess of the calculations, could not be docked. She was docked some time ago it seems, however, at the Brooklyn Navy Yard by being lightened of practically everything movable, and taking advantage of an unusually high tide which comes once a month over the sill of the dock, provided there are favorable winds at the proper time. Only at this time can the cruiser be floated in or out, and from this it follows that she may at any time be caught in a trap and made useless for a month or two.

One of the lessons which come by experience was taught some painters in Belvidere, N. J., the other day. They were painting the interior of a stand pipe of the waterworks and were using paint mixed with benzine. Being dark in the pipe they had lights to work by, and the air within the pipe becoming mixed with benzine vapor an explosion took place, and the pipe was immediately filled with flame. We judge from the report that the fire was mostly above the men, and hastily letting themselves down to the bottom they passed out and escaped with the loss of their paraphernalia only.

The secretary had been showing a mechanical friend through the works, when the latter remarked: "I see you have some very nice pneumatic hoists here."

"Oh, no," replied Mr. Secretary, "they work with air."

The London Times concludes that "The confidence of capital has been rudely shaken, and can only be revived by decisive evidence of a complete change in the method of railroad management in the United States."

LETTERS FROM PRACTICAL MEN.

A Wonderful Cupola.

Editor American Machinist:

Under the above heading an article appeared in the American Machinist September 27th, showing a cupola supposed to be doing phenomenal work, but upon a careful consideration and comparison with other cupolas on the same basis, a great deal of the wonderful disappears. The economy is not as great as supposed at a casual reading.

In the first place the ratio of fuel to iron is given on the basis of what is used between charges, leaving the bed out entirely, neglecting the fact that the bed is a most important factor in raising the temperature of the cupola and all it contains. At the same time the first charge of iron is melted on the bed, and is clear gain in ratio of fuel on this basis. No charge being made against it a considerable alteration is made in the proportion, especially if the heats only run from one to two hours.

From my own experience I can assert that, based on the preceding basis, I have accomplished an equal result with that shown by Mr. Bollinckx.

The past month I have used a cupola built on the plan shown in the American Machinist, September 20th, 30 inches in diameter clear of the brick. Aside from the bed I have used fuel at the rate of 1 to 20, taking the coke and iron relative in each charge, but upon the addition of the charge melted on the bed the ratio will run up to 1 to 22 and 1 to 23; but taking the same heats and adding in the fuel used on the bed the ratio is brought down to the more rational figures of 1 to 14.

I can safely say I have not reached the limit of this cupola, but my experiments have been delayed on account of several cars of poor coke. I had not intended to write of this cupola at present, and later on will present a paper on it, but American pride bade me proclaim what is done in our country when its reputation for excellence is at stake. The iron used was from 30 to 50 per cent. pig iron, and the remainder heavy railroad scrap.

The cupola melts from 4½ to 5 tons per hour. I make no special claims as to the economy of the cupola, but rather to the management.

As in the case of the Greiner, there is no flame shown at the charging door, and very little gas or smoke, the temperature at that point not running over 125° Fahr., but this is not due to the construction of the same, but the manner of charging, and any cupola can be handled so that has from 9 to 11 feet height to the charging door.

Any further information desired on this subject I will cheerfully give if in my power through the columns of the American Machinist or by personal letter.

Herbert M. Ramp.

Springfield, Mo.

"Our Glorious Navy."

Editor American Machinist:

I desire most heartily to echo the hope expressed in your editorial of September 27th under the above caption, "that some day a man, perhaps an engineer, or an experienced manager of industrial affairs elected to Congress, will bring about a thorough investigation of naval matters and an entire reorganization of the navy upon the basis of common sense."

As an old-time member of the engineer corps of the navy, for nearly thirty years now past in private life, I have never lost that interest in the service which an active part in it for eight years engendered; only to note, however, continuously, and up to this late day, undiminished by progress or time, the same ignorant spirit of selfish unfairness and mean persecution of the staff corps, and particularly the engineer corps, on the part of the line, which existed thirty years ago, and which was the primary and principal cause of the writer's resignation therefrom in 1869.

You characterize aright the action of the Navy Department in Engineer Danforth's case as nothing more or less than an outrage, but it is no greater outrage than that of Assistant Engineer Bennett and hundreds of other cases varying in kind and importance which have come under the writer's notice.

In the development of the modern naval vessel, now—aside from its navigation and artillery aspects—a vast aggregation of steam and electric machinery, the logic of progress demands that the marine engineer should occupy a correspondingly higher position on such vessels.

The fact is that the line of the navy have been endeavoring for many years to hold the service down to the dead past. When the motive power of ships was had through the instrumentality of sails, the present type of naval line officer was an appropriate one in all respects, but since the advent of steam, a new class of executive men, both on shipboard and ashore, has become a necessity. The profession of old, known under the name of seamanship, *i. e.*, the management and propulsion of the ship by means of sails, has passed away, and the endeavor to retain with the new fighting machines the traditional seaman is no better than a harmful anachronism. In his place should be largely found the engineer in his more exacting, more difficult and more scientific branch of the service, marine engineering. It is impossible longer to belittle the importance of the latter in the management of the modern warship. Common sense teaches us that with ships in which such a thing as a sail is unknown, but which is little less—aside from its guns—than a steam machine from stem to stern, the number of engineer officers, and their appropriate status thereon, should have kept pace with the changes in the character of the vessels; while what is now known as the line could and should have been correspondingly reduced. But, instead of this, we find line officers not only on shipboard, but on shore, everywhere appropriating the legitimate duties and prerogatives of the engineer, and doing everything possible to suppress and emasculate him as a part of the naval personnel.

It is not too much to say that the chief engineer of a modern war vessel should be second only to the commanding officer, both in rank and command, and that he should have undisputed authority over his men and his department, subject only to the commander of the vessel. It is equally patent that there should be fewer subordinate line officers, and more subordinate engineer officers on such ships.

What can be more absurd than the delegation of young line officers as inspectors of materials of construction of ships, machinery, armor or even guns, displacing men of the engineer corps who have been especially trained for such work, and who by all rules of common sense must be better equipped for such duty. But this is done to find places for a redundant line corps, and with what detriment recent results have too plainly shown.

All this, however, is but a part of the traditional and insensate jealousy and petty persecution of engineers which has always characterized the average line officer since the advent of steam in the navy.

If the officers of the engineer corps were guilty of half the inefficiency repeatedly shown by the line in the discharge, even, of their legitimate duties, the country would never hear the last of it. The professional shortcomings of the line are notorious to those who are in a position to know of them, and this is just what should be looked for where they expend so much of their energy in trying to perform the legitimate duties of others. But to outsiders these shortcomings are always conveniently hidden from view, if not officially hushed up. I have no doubt but that Mr. Harrison's estimate of the cause of the loss of the "Chesapeake" is a just one, and that the loss of the "Kearsarge" was due to similar causes.

The writer has assisted at the making of a little naval history of this kind: When the flotilla under Melancthon Smith—then a captain, I believe—went into Albemarle Sound, with the purpose of destroying or capturing the ram "Albemarle," and retaking Plymouth, N. C., the writer was chief engineer of the flagship, the "Mattabessett," one of the four or five "double enders" which were a part of the U S fleet. Upon our arrival there the squadron was running short of coal; coal schooners being expected daily. Lying at anchor some miles below the mouth of the Roanoke River, word was passed through the fleet to be as economical of coal as possible, in view of the fact that the ram might appear before the arrival of the expected coal. This was a judicious and proper order. At the same time the chief engineer of every vessel was under previous orders to keep the machinery in readiness to go ahead at full speed at a moment's notice. This was all right and proper, and put responsibilities just where they belonged. But the commanding officer of the "Mattabessett"—now a commodore or rear admiral—felt called upon to exercise some superior authority; and sent me an absurd order to confine myself to a certain consumption of coal per day. The quantity was so absurdly small that in the merest self defense I sent a written reply to the captain that it would be impossible to comply with both that order and the previous one to have the ship ready to move at full speed. The last order was insisted on without explanation or consideration of any other condition than that he was captain, and proposed to be obeyed. Of course, I obeyed the last order. Result: When the ram appeared the flagship was disgraced by not being able to keep up with the remainder of the ships, on none of which had this absurd order been promulgated; arriving at the scene of the engagement some half hour behind the rest—although in time for the commencement of the fight. As this overwise captain had to make some show of putting the blame on some one, I was—after the fight with the ram was over—suspended from duty by him. I demanded a court-martial, but was accorded that convenient naval instrument for the escape of mistaken or evil minded accusants, a court of inquiry. I pleaded my own case, and was acquitted of all blame. But what of the real culprit—the knew it all commander? Was he court-martialed, or brought to account in any way? Oh, no; he was of the immaculate line, and, like the king, could do no wrong; so nothing more was said or done about it.

This is a fair sample of thousands of cases which make the personnel of the navy what it is, which engender and keep alive what has so long been notorious as the line and staff fight, and which interferes more greatly with the discipline and efficiency of the service than civilians have any idea of.

I call in mind another sample of this kind of naval history quite characteristic of the inefficiency of the line officer of that day. In January, 1862, the sloop of war, "Pensacola," left Hampton Roads bound for Key West and Ship Island preparatory to joining Farragut's fleet for the taking of New Orleans. The ship was fitted with machinery which at that time was no better than a very risky experiment, and it proved the severest task upon the engines of the ship to keep them going at all. I quote from a diary which I kept at the time:

"Monday, Feb. 3, 1862.

"Beautiful day; steamed continuously. Went on watch at 8 P. M. In consequence of the illness of several senior engineers, owing to extreme heat of engine and fire-rooms, I was entrusted with charge of the engine department for the first time. Was, during the evening, repeatedly requested by the chief engineer to increase speed of engines if possible. Was doing my best endeavors with some success, when, at 11 30 P. M., while making about 10½ knots, the vessel struck violently and ran through an opening in a coral reef, known as Carysfort reef, a part of Elliot's key, on the coast of Florida. After an hour's endeavor—to get her off—with considerable confusion on deck, it was relinquished for the night. Fired minute guns, and made signals of distress. There were two lighthouses in sight just before running on the reef, and one as we lay there, but just what they were not one of our profound navigators could seem to determine. If they had known, I suppose we would not have run ashore."

Here was a clear, starlight night, and two lighthouses in sight, yet these "profound navigators" deliberately ran the ship square on shore. One would think that if they could not interpret the lights they might at least have found from the stars or compass that in that vicinity they could not have expected to steer the course they did without sooner or later fetching up somewhere in the State of Florida. As it was dead high tide when she went on, we lay there for 10 days in a Providential calm, until the ship was stripped of everything ponderable, from her lower mast-heads to the keelson, before she could again be gotten into deep water.

Now, suppose this ship had drifted or gone ashore because of some shortcomings on the part of the engineers: is it to be supposed that they would not have been held to a strict accountability? Yet, so far as the writer was ever able to learn, not even a court-martial came of it. No one responsible for the disaster ever suffered so much as a reprimand. The ship went to New Orleans, did her part in the great fight at the forts below, and the whole matter was hushed up and forgotten, and the then executive officer—so called—of the ship is now a rear admiral, I believe.

It is time the American people awoke to a realizing sense of the inordinate vanity and unrepublican spirit manifested by the line of the navy, and that the man or engineer be elected to Congress who will "bring about a reorganization upon a basis of common sense."

The present Assistant Secretary of the Navy, in an extended article in the *North American Review* for October, on the "Reorganization of the Personnel of the Navy," enters into an exhaustive showing of what should be done to equalize and hasten promotion in the line, and even enlarges upon the great necessity for promotion for warrant officers, and the special need for consideration of the apprentice system; but he dismisses the most grievous wrong which obtains in the service, and which has for many years been growing worse instead of better, with this: "Leaving out what may be considered injustices in the staff and auxiliary corps," and "the demands of the staff officers, and the frictions and contentions which, like warring creeds, have at times existed between them and the line, have not been considered in this article, because it has been deemed best to center attention upon a serious abuse which cannot much longer await a remedy," namely, the more rapid promotion of the line. He finds in the line part of the navy register an objectionable "hump," very difficult to smooth down. Let him cut it out and supply its place with much needed engineers, better equipped to perform a large part of the duties now put in the hands of junior officers of the line, by men who should have a larger part in the management of the steel, steam fighting machines which they design and build.

John T. Hawkins,
Late Past Asst. Engineer, U. S. N.

Taunton, Mass.

Mysterious Leaks of Compressed Air.

Editor American Machinist:

Your genial correspondent, "Quirk," whose remarks we all read with pleasure, and some of us with considerable amusement, asserts, or rather re-asserts, October 4th, that "there are mysterious disappearances of compressed air," while just above that he asserts that "machinists, as a rule, are not superstitious."

I was thinking lately of the many successful applications of compressed air, and of how they might be arranged in groups, each group depending upon some peculiar property which air possesses, and which can be found nowhere else. Curiously enough, I had thought of one quite important group that depends for its eminent success upon the reliability of the air. It is entrusted

with the most responsible duties of the watchman or the signalman, simply because it is pre-eminently to be depended upon, and is not given to mysterious disappearances. It stands on duty night and day without loss and without cost until it is time for it to act, then it acts at once. Where does the mysterious disappearance theory come in, and where are the undiscoverable and incurable leaks, in the switch and signal service upon our railroads, where pipes are run for twenty miles? In the air-brake service, where cars are so quickly and apparently so carelessly, coupled, and where everything is subject to such incessant and violent rattling we should surely find incurable leaks if anywhere, but who ever heard of them? Who would equip costly buildings with automatic sprinklers operated by compressed air, if the air were apt to mysteriously disappear about the time that a fire occurred?

So many establishments are now equipped with a compressed air supply, especially for hoisting purposes, or for operating isolated machines, or machines that are intermittently employed, the pipes ramifying through extensive buildings and yards, the air pressure constantly maintained, and the whole accomplished in every case with such satisfaction and profit that it seems to be hardly the correct thing to be harping about incurable, undiscoverable leaks or mysterious disappearances of the air, and I think I may reiterate my assertion that "there are no losses through leakage that are not as easily preventable as steam leakages, and by the same means," and the losses from this source, under the supervision of skilled and faithful mechanics, in whose brains there is no "mysterious disappearance," maggot crawling, are not worthy to be mentioned or thought of in comparison with the losses through steam condensation.

One thing that all may not think of, and which I venture to submit to the consideration of Mr. Oberlin Smith, is that the conditions of leakage and of curing the leaks in air pipes are very different from the conditions affecting steam pipes. Air pipes, as a rule, that are once tight will remain tight. They are not subject to the expansion and contraction strains that play the mischief with steam pipes, and they also know nothing about water hammer. Who wants to say and maintain in the AMERICAN MACHINIST that we cannot run a line of pipe that will be practically without leak? The facts show that for compressed air it can be done, has been done, is being done.

FRANK RICHARDS.

Formula for Unstayed Plates.

Editor American Machinist:

Referring to Mr. Albert E. Guy's letter in your issue of September 27th, in reply to my criticism of his formula for thickness of unstayed circular plates fixed at the edges, I have to acknowledge the error which he points out in the formula I gave for maximum bending moment in a beam uniformly loaded. It should be $\frac{1}{12}\ W\ l$ instead of $\frac{1}{24}\ W\ l$. The error arose from my taking the figures $\frac{1}{24}$ from the next line below Professor Wood's equation No. 126 (second addition page 127), instead of the $\frac{1}{12}$ which appears in the formula itself. The discovery of this error destroys that part of my criticism based upon the hypothetical case of the plate being sawn in three pieces, as far as the uniformly loaded plate is concerned. It does not, however, affect my criticism of the theoretical basis of Mr. Guy's formula, upon which I still insist. If the statement that the tensile strain is a maximum at the point of application of the load (which is inaccurate in the case of a beam fixed at the end and uniformly loaded) be omitted, it by no means weakens the argument against Mr. Guy's theory, which shows that it is based on an incorrect hypothesis. This argument Mr. Guy has not answered.

Although the hypothetical case given in my last letter failed to show the error of Mr. Guy's formula, I think it may be shown by two other similar cases which I give below.

Mr. Guy's formula for thickness of circular plates fixed at the edge and loaded uniformly is $h = 0\ 938\ r \sqrt{\frac{p}{f}}$, in which h is the thickness of the plate, r the radius, p the load per unit of area, and f the coefficient of safe resistance of the material to transverse strain.

If we let W = the total load on the plate, $W = \pi\ r^2\ p$; $p = .3183\ \frac{W}{r^2}$, and the formula reduces to $h = .5294 \sqrt{\frac{W}{f}}$.

Grashof's formula, as given by Unwin, when reduced to the same form, gives, $h = .460 \sqrt{\frac{W}{f}} = 816\ r \sqrt{\frac{p}{f}}$.

In Fig. 1 let the dotted circle represent the inner edge of the circular ring to which the plate is firmly fixed, as by being held down by a ring of similar size above the plate, the two rings and the plate being all bolted together. The dotted circle represents also the loaded surface of the plate. Draw an inscribed square on the plate as shown by the shaded area and conceive that the whole load instead of being distributed over the area of the circle, is confined to the square. The area of the square is $2\ r^2$, or less than two-thirds of the area of the circle; its side is $1.4142\ r$, and the versed sine of each of the segments from which the load is removed is $(2 - 1.4142)\ r \div 2 = .2929\ r$.

Now remove the supporting rings and support the plate by four straight supports that are tangent to the circle, as shown in the figure; next, cut away four right angled corners out of the plate as at a, d, e and f, the corners touching the circle, so that the plate becomes a cross of four arms, each of which has a width of $1.4142\ r$. We have now done three things which should tend to diminish the resistance of the plate and which should make it necessary to increase its thickness, if it is to support with the same factor of safety the same total load, viz: 1, we have concentrated the load into a space of two-thirds of the space it originally occupied; 2, we have removed the points of support back from the original ring to the line of the straight support; and 3, we have cut out four corners which by their resistance to buckling would increase the resistance to bending along the lines $d\ a$, $a\ b$, etc. If now we should calculate the thickness required at the lines $d\ a$, $a\ b$, etc., to resist bending, using the same factor f for working strength of the material, and should find that even less thickness was required under these unfavorable conditions than the thickness of the plate given by Mr. Guy's formula, it is only reasonable to conclude that there is an error in the formula.

The total load being W, the reaction of each of the four supports is $\frac{1}{4}\ W$, and the moment of the reaction of the support $e\ f$ about the line $a\ b$ is $a\ c \times \frac{1}{4}\ W = .2929\ r \times \frac{1}{4}\ W = .0732\ W\ r$.

The moment of the resistance of the plate at $a\ b$ is $\frac{1}{6}\ f \times 1.14142\ r \times h^2 = .2357\ f\ r\ h^2$.

Whence $.0732\ W = .2357\ f\ h^2$, and $h = \sqrt{\frac{.0732\ W}{.2357\ f}} = .557 \sqrt{\frac{W}{f}}$ for a plate resting on the supports, but not firmly fixed thereto. The maximum stress in a beam fixed at the end and uniformly loaded, being two thirds of that in a beam merely supported (while in the case of a beam loaded in the middle it is only one half as great) it is more than safe to take the factor $\frac{2}{3}$ in this case, and then $h = \sqrt{\frac{2}{3}} \times \frac{.0732\ W}{.2357\ f} = .455 \sqrt{\frac{W}{f}} = .803\ r \sqrt{\frac{p}{f}}$ as against $.938\ r \sqrt{\frac{p}{f}}$, which is found by Mr. Guy for a much more favorable case.

Second Case.—In Fig. 2 let the dotted circle represent the supported edge of the circular ring whose radius is r, and the shaded portion an area in which the load is concentrated, the distance from the inner edge of the ring to the edge of the load being represented by $c\ r$ in which c is a decimal fraction.

Conceive that thin radial saw cuts are made from the outer edge of the plate up to the margin of the loaded area, dividing the supporting portion of the plate into a great number of radial bars. Let the number of these bars $= n$, then the reaction of the support upon each $= W \div n$, and the moment of reaction considered with reference to the margin of the loaded area $= W\ c\ r \div n$.

The circumference of the loaded area is $2\ \pi\ (r - c\ r) = 2\ \pi\ r\ (1 - c)$, and the portion of it occupied by the inner end of one of the radial bars is $2\ \pi\ r\ (1 - c) \div n$. The moment of the bending resistance at that point is $\frac{1}{6}\ f\ 2\ \pi\ r\ (1 - c)\ h^2 \div n = W\ c\ r \div n$.

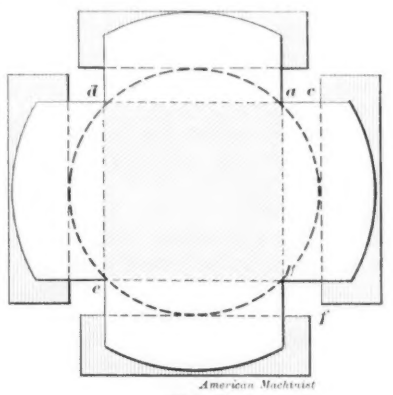


Fig. 1

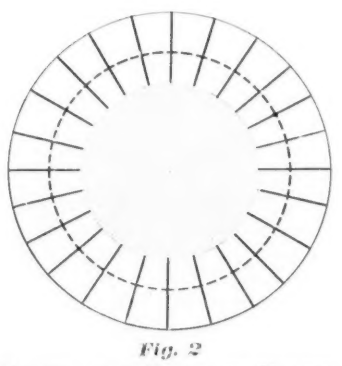


Fig. 2

FORMULA FOR UNSTAYED PLATES.

Whence $1.0472\ f\ (1 - c)\ h^2 = W\ c$, and $h = .977 \sqrt{\frac{W}{f} \frac{c}{(1 - c)}}$; this is for the plate supported but not fixed. Multiplying by $\sqrt{\frac{3}{2}}$ for a plate fixed at the edges we have

$$h = .798 \sqrt{\frac{W}{f} \frac{c}{(1 - c)}} = 1\ 414\ r \sqrt{\frac{p}{f} \frac{c}{(1 - c)}}.$$

If $c = .1$, $c \div (1 - c) = \frac{1}{9}$; $\sqrt{\frac{1}{9}} = \frac{1}{3}$;

$$h = .266 \sqrt{\frac{W}{f}} = 471\ r \sqrt{\frac{p}{f}}$$

If $c = .2$, $c \div (1 - c) = \frac{1}{4}$; $\sqrt{\frac{1}{4}} = \frac{1}{2}$;

$$h = .398 \sqrt{\frac{W}{f}} = 707\ r \sqrt{\frac{p}{f}}.$$

If $c = .3$ $c \div (1 - c) = \frac{3}{7}$; $\sqrt{\frac{3}{7}} = 655$;

$$h = .523 \sqrt{\frac{W}{f}} = 926\ r \sqrt{\frac{p}{f}};$$

that is we would have to push the load in from the edge of the plate to a circle only seven-tenths of the diameter of the plate or less than one half the area and run the radial saw cuts in three tenths of the radius all around before we would establish a condition requiring even for these disconnected bars a thickness equal to that given by Mr. Guy for a solid plate.

The hypothetical cases, however, I consider as merely cumulative evidence against Mr. Guy's formula; the chief objection to it is that it is based on an untenable theory, viz.: that a circular plate fixed at the edge and loaded uniformly bears a likeness to a beam of uniform thickness and triangular in plan loaded at the apex and fixed at the base.

Mr. Guy asks a few questions in his letter which seem to call for categorical answers.

1. "What will happen if the plate is bored concentrically and becomes a flat ring held at the outer edge and loaded along its inner edge?" I do not know. I know of no experiments on such a case. It would be extremely difficult to solve such a problem on a theoretical basis. I do not believe, however, it would act as a trapezoidal beam loaded at its smaller parallel end, which is free to move, and fixed at the other, because the continuity of the inner edge of the ring prevents its free movement. A load causing a tendency to bend the ring would develop a tensile strain in a circumferential direction in the inner edge.

2. "Will not the bending moment vary from zero at the inner edge to a maximum at the outer?" No, not as long as there is any tensile resistance in the direction of the inner circumference of the ring. If this resistance could be destroyed and the ring become a series of disconnected trapezoidal beams then the moments would vary as stated.

3. "By reducing the inner radius to zero (infinity, it reads in the original, probably a slip of the pen) will not the plate be full again and subject to the same laws that govern the flat ring?" Yes, the plate will be full again, but no, it will not be subject to the same laws as the flat ring; for the full plate has a continuity of metal across a whole diameter, and the metal at the middle of the diameter will be subject to a compressive stress on the side of the load or upper side and to tensile stress on the other side, while, if instead of the continuity of metal there is a hole there can be no stress in the hole and the laws of the flat ring cannot apply. To analyze the problem of a flat plate on the hypothesis that it is a circular ring with an inner radius of zero is almost as absurd as if we should analyze the problem of a beam fixed at the ends and uniformly loaded by considering it as a special case of two beams, each fixed at one end, in opposite abutments, in line with and projecting towards each other, with the distance between their free ends reduced to zero, and the ends fastened together by a connection whose strength is the same as that of a solid beam.

Mr. Guy deserves credit for his attempt to solve the difficult problem of obtaining a satisfactory formula for flat plates, although, as I believe, he has not as yet solved it. I trust he will give us a translation of the results of Professor Bach's experiments which he mentions and show wherein they tend to confirm his formula.

Unwin, "Elements of Machine Design," eighth edition, page 76, says that some experiments by Mr. R. Wilson (*Engineering*, Vol. XXIV., p. 239) appear to show that the ultimate resistance of flat plates is considerably greater than that obtained by Grashof's formulæ. Since Mr. Guy's formula gives thicknesses more than 12 per cent greater than Grashof's, the results of Mr. Wilson's experiments would be farther away from those calculated by Mr. Guy's formula than from those calculated by Grashof's. It will be interesting to learn if Professor Bach's experiments show a closer agreement.

WM KENT.

Passaic, N. J.

Fraudulent Testing.

Editor American Machinist:

The effect of rapid and slow testing was clearly shown a number of years ago and repeatedly since. It is, in the first place, an admitted fact that results are higher when testing very rapidly and using a multiple lever knife edge machine. A number of pieces were cut from a bar of Bessemer steel of about 70,000 pounds tenacity, and while six were tested slowly on an emery machine, six others were tested as rapidly as the skill of operator permitted. The results showed *no definite differences*. The reports of the Bauschinger conferences also contain a statement, that "for the usual speeds of testing in use, no appreciable differences in resistance were noticed." These facts distinctly show that there are differences due to type of machine, and I think that your correspondent "C." knows what he was writing about.

GUS. C. HENNING.

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Some Phases of the Machinery Business—Foreign Trade.

The present prevailing dullness amongst machinery builders presents some curious phases, especially in what may be called its distribution. If one goes among certain classes of machinery builders and dealers he will hear practically nothing but the most discouraging reports as to present business and the prospects of an improvement; while at the same time others are quite busy, have all the orders they can conveniently take care of, and look to the future from the most cheerful—even optimistic point of view.

This unequal distribution of business is sometimes observable even in a single shop, where the business is somewhat varied, a very prominent machine tool establishment being at the present time obliged to run some departments double time with two sets of men, while other departments are so dull as to be shutting down at 4 P. M.

All those whose business depends largely upon railroads find things very dull, as the railroads are generally doing as little in the way of repairs and renewals as is consistent with keeping the roads in operation; and as the railroads, under normal conditions, absorb such a very large proportion of the regular product of machinery, their practical withdrawal from the machinery market is very seriously felt, and will probably continue to be the dominant influence until some turn of affairs in Wall street, heavy crops in the West, or sheer wearing out and practical disappearance of equipment compels renewal; and those who are most familiar with practical railroad affairs declare that this latter condition is approaching on some roads, and must soon make its influence felt by "supply men" and machinists.

On the other hand, those who depend for business mainly upon the development of electrical matters are finding trade fairly good, and some are very busy, though there is much complaint of low prices, and these will probably continue until productive capacity is more nearly balanced by demand. One engine builder informed us the other day that he was exceedingly busy, and had been all summer—was now running his shop at night and employing a double shift of men. Not only this, but he declared that nearly all the engine builders who pushed things a little were finding business at least fairly good, while a number of them were quite busy.

The general aspect of the machine tool industry is not especially cheering. For reasons well understood, machine tool builders are usually first to feel a decline in business and last to feel the influence of a return of prosperity, because business must revive sufficiently to bring practically all the existing equipment of machine tools into use before there can be a general demand for additional tools.

This, however, does not apply to those tool builders who have something in the way of a specialty designed to reduce the cost of production as compared with standard tools and methods. Most of these are finding plenty to do by simply letting people know what can be accomplished by the adoption of their methods and machines.

As indicating the very wide field and the multitude of opportunities for putting in electric installations, including engines and boilers, an engine builder recently put into a single apartment house, in this city, four engines aggregating 400 horse-power, and in another case of a large building in a nearby city, which had been heated by steam and lighted by gas, the boiler plant was improved and fitted for higher pressure, engines and dynamos put in for lighting, and, by using the exhaust steam for heating, both heating and lighting is now done on a smaller consumption of coal than was formerly required for heating alone. Of course, in this case the work was done under the direction of a competent engineer; which is always an important requisite where the best results are wanted, but when such results are possible it indicates that there are many places where engines ought to and probably will go in soon.

In this connection should also be mentioned the fact that those machine builders who have, in recent times, taken some pains to cultivate foreign trade have generally been in much the best condition during the dull times, some of them finding their foreign business nearly, if not quite, equal in volume to the home trade.

American machine tools, and woodworking machinery especially, are making rapid headway in England and on the Continent, and the interest taken there in our machinery and methods is plainly shown by its effect upon the foreign circulation of this journal, and the constantly increasing number of inquiries pertaining to American machinery received by us and by those whose machines are illustrated or otherwise represented in our columns. This matter of foreign trade in machinery is, we think, one of very considerable importance, and is something which should be carefully looked into by every machinery builder who has not already done so.

Small Caliber Shoulder Arms.

Recent developments in infantry and naval small arms afford an interesting example of what in mechanics would be called mechanical equivalents, and the result as it now stands is founded upon the same general principle by which an engine with a very small cylinder, but driven at a very high speed and at high pressure, may do as much or more work in a given time than can be done by a much larger engine working at slow piston speed and with low pressure steam.

The caliber adopted for the new navy magazine gun is smaller than any previously adopted by any military power, and as this caliber, .236 inch, has been adopted as the result of careful observation and study of the effect of previous reductions in our own and other armies, it is probable that calibers will remain small at least until there comes some discovery or improvement of sufficient importance to bring about a revolution in small arms construction based upon new principles.

When the isolated fact that the bullet of the new naval arm will be appreciably less than one fourth of an inch diameter is considered, it seems impossible that such results can be attained as are expected from it. It seems too much like using a squirrel gun in a tiger hunt; but the science of mechanics, and experience as well, teach that as much energy or "stored work" can be put into such a small diameter bullet as into a larger one if only it is made long enough to give it the required weight, and enough powder is effectively burned behind it to give it the required velocity; and with an equal amount of stored energy the smaller bullet has the advantage of less area opposed to the air, a flatter trajectory and deeper penetration as a result of higher velocity and smaller resistance of the substance struck.

No one, we believe, yet knows the smallest practicable caliber that can be employed to advantage, but there are experts that believe that .216 inch can be used, and, besides the other reasons for constantly reducing calibers to the smallest diameters believed to be practicable, there is the important consideration of reduction of weight of the arm and increase in the number of cartridges that can be carried by its user, an object to which the invention of modern high pressure powders burned in an enlarged powder chamber contributes much.

An unusual case is likely soon to be decided by the U. S. court at Philadelphia where some employes of the Reading Railroad have asked for an injunction preventing the receivers of that road from discharging them for belonging to an organization.

We think this tendency to resort to the courts in the settlement of disputed points that can be settled in no other way is a good tendency. It amounts simply to finding out what the law is upon these points, and then if the law is found to be unjust, or for any reason unsatisfactory, the sovereign people can change it if they so decide.

It is our belief, as we have said before, that it is wrong in principle and extremely impolitic in practice for any employer to attempt to say that no one employed by him shall belong to any given organization. When an employe or an organization wrongs an employer then it is time enough for that employer to act, and previous fair treatment of them makes the victory all the more certain where the employer is clearly right. But in the case of these Reading employes it is, after all, difficult to see how they can be protected because so many other causes can be given for discharging a man where there is a desire to get rid of him.

Literary Notes.

In the October number of *The Engineering Magazine*, a new department is opened called the "Review of the Industrial Press." In this it is proposed to notice and review the more important articles appearing in the various technical publications, and, in short, to do for industrial literature about what *The Review of Reviews* does in the political and literary field.

The volume of technical literature now appearing in this country and abroad is very great, and when all branches of engineering are included, as we suppose is the intention, it becomes evident that an adequate presentation of the best or leading ideas will involve not only a great deal of labor, but also, it seems to us, necessarily result in gathering together in one publication a very considerable mass of matter, which cannot, in the nature of things, be interesting or useful to very many readers; and specialists, which is what most men nowadays are, will still have to depend upon publications devoted to their specialties. Nevertheless, we shall hope to see this ambitious undertaking succeed, and the fact that industrial literature has grown to such proportions as to tempt any one to undertake such a work is significant.

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Engineers will find this book to be exceedingly handy and useful. All the data and tables needed for computations for boiler testing, duty performance of pumping engines, and for indicator diagrams in general, are given in compact form. The research, the computations of tables, and the comparisons, made by the author should be a guarantee for the accuracy and reliability of the work. D. Van Nostrand Co., 23 Murray and 27 Warren streets, are the agents for the United States. Price of the book $2.

QUESTIONS AND ANSWERS.

Questions of general interest relating to subjects discussed in our columns will receive attention in this department. The writer's name and address should always accompany the question. Neither correct initials nor location will be published when there is a request to that effect. If questions are enclosed with a business communication, they should be written on a separate sheet.

(484) W. M. M., ——, asks for a metal which can be poured into a bearing about a shaft and which will expand in cooling in such a way as to give the required amount of clearance to the shaft. We know of no such metal which would be suitable for journal bearings, but, perhaps, some of our readers can give the desired information.

(485) R. E. S., Bethlehem, Pa., writes: Kindly send me as many references as you can on the the subject of "oil as fuel,'

other than those given in the index of the AMERICAN MACHINIST, which I have already on hand. *A.*—The book treating on fuels, solid, liquid and gaseous, their analysis and valuation, by H. J. Phillips, is a good one, and may be of aid to you. Outside of this book and the articles published in our columns, we cannot call to mind any other references.

(486) J. C. B., Brooklyn, N. Y., writes: Kindly inform me how to tin by dipping small articles made of sheet iron or steel. I have nearly succeeded, but there is something wrong; they are too rough. *A.*—The articles must be cleaned with sand, and, if necessary, with acid, and then washed in water. When cleaned, they are put into a bath prepared with cream of tartar 1 ounce, protochloride of tin 1 ounce, water 10 quarts. This bath should be kept in a wooden or stoneware vessel at a temperature of 190 degrees. Small pieces of zinc should be distributed among the articles in the bath. When the coat is considered thick enough, the articles are taken out of the fluid, washed with water and dried.

(487) K. B., New York, writes: You will help me very much by giving me the formulæ for computing the thickness of cylinders subjected to internal pressures, and apply them to the following problem: What should be the thickness of a cylinder 10 inches diameter inside, to stand a pressure of 2,500 atmospheres, it is to be 4 feet long and made of cast steel; will it not have to be built up like a gun; that is, one cylinder shrunk upon the other? *A.*—There are several formulæ for computing the thickness of cylinders, depending on the ratio of thickness to diameter. When this ratio is very great, as is found in boiler shells subjected to an internal steam pressure, the thickness is found by the following formula:

$$t = \frac{p \times r}{s}, \qquad (1)$$

in which t = thickness in inches, p = pressure per square inch, r = inner radius of cylinder, and s = the stress per square inch in the material. For very high pressures, as given in your example, the following formula is employed:

$$t = \frac{p \times r}{s - p}, \qquad (2)$$

in which the letters denote the same quantities as before. Assuming that the pressure you give is the pressure per square inch, we have for p $2{,}500 \times 14\ 7 = 36{,}750$ pounds. For the ultimate tensile strength of cast steel we may take 86,000 pounds per square inch. Now substituting for the letters in formula (2) their values, we have

$$t = \frac{36{,}750 \times 5}{86{,}000 - 36{,}750} = 3.73$$

inches for the thickness of the cylinder at which the cylinder will burst. If we take 43,000 pounds per square inch for the safe value of s, we have

$$t = \frac{36{,}750 \times 5}{43{,}000 - 36{,}750} = 29.4 \text{ inches}$$

for the thickness of cylinder walls. From formula (2) we obtain the following:

$$p = \frac{s \times t}{r + t} \qquad (3)$$

from which we can compute the internal pressure any cylinder will stand when the diameter and thickness of wall is given. Substituting in formula (3) for the letters their values as previously assumed we have

$$p = \frac{43{,}000 \times 29\ 4}{5 + 29\ 4} = 36{,}750 \text{ pounds}$$

per square inch for the internal pressure, as we might have expected. In ordnance construction the following formula for finding the internal pressure of cylinders is frequently used:

$$p = \frac{3\,(D^3 - d^3)\,s}{4\,D^3 + 2\,d^3}, \qquad (4)$$

in which D = external diameter in inches, d = internal diameter in inches, the other letters denote the same quantities as before. If we again adopt a value of 43 000 pounds per square inch for s, and adopt the thickness of 29 4 inches for the cylinder walls which will make $D = 10 + 29\ 4 + 29\ 4 = 68.8$ inches; d, is, of course, 10 inches, and substituting these values for the letters in formula (4) we have

$$p = \frac{3 \times (68\ 8^3 - 10^3) \times 43{,}000}{4 \times 68\ 8^3 + 2 \times 10^3} = 31{,}238$$

pounds per square inch for the internal pressure. This shows that the thickness 29 4 inches of the cylinder walls as found by formula (2) is scarcely large enough. Professor Reuleaux, in his "Constructor," gives Lame's formula for finding the thickness of the cylinder walls as follows:

$$t = \left(\sqrt{\frac{s+p}{s-p}} - 1\right) r \qquad (5)$$

in which the letters denote the same quantities as in the other formulæ. Adopting for s a value of 43,000 pounds as before and substituting the values of the letters in formula (5), we have

$$t = \left(\sqrt{\frac{43{,}000 + 36\ 750}{43\ 000 - 36{,}750}} - 1\right) \times 5 =$$

11 35 inches for the thickness of cylinder walls. But, according to formula (4) this thickness is not large enough for the pressure; hence we should adopt the thickness as found by formula (2) for a cylinder cast with solid walls. When a cylinder is built up by shrinking one shell over another, the strength of the walls is much increased, and consequently their thickness can be reduced. The formulæ and methods employed in compound or built-up construction of cylinders are complex, and may be found in modern works on ordnance.

(488) G. K., New Orleans, La., writes: The inclosed sketch, Fig. 1, represents a wrought iron crane. What I would like to know is how to compute the strains in the different members so as to determine their dimensions. The crane is to support a weight of 1,000 pounds as shown. *A.*—First draw the center lines through all the members and thus obtain a skeleton plan, as shown in Fig. 2. We may consider the horizontal jib $E\,L$ to be a beam in equilibrium supported at B and loaded at each end. We now compute the vertical pressure at B due to the loads on the lever $E\,L$. This is found by the principle of the lever to be $\frac{42 \times 1{,}000}{26} = 1{,}615$ pounds, fraction neglected. Draw the vertical line $B\,C$, say, 8½ inches long, and through C

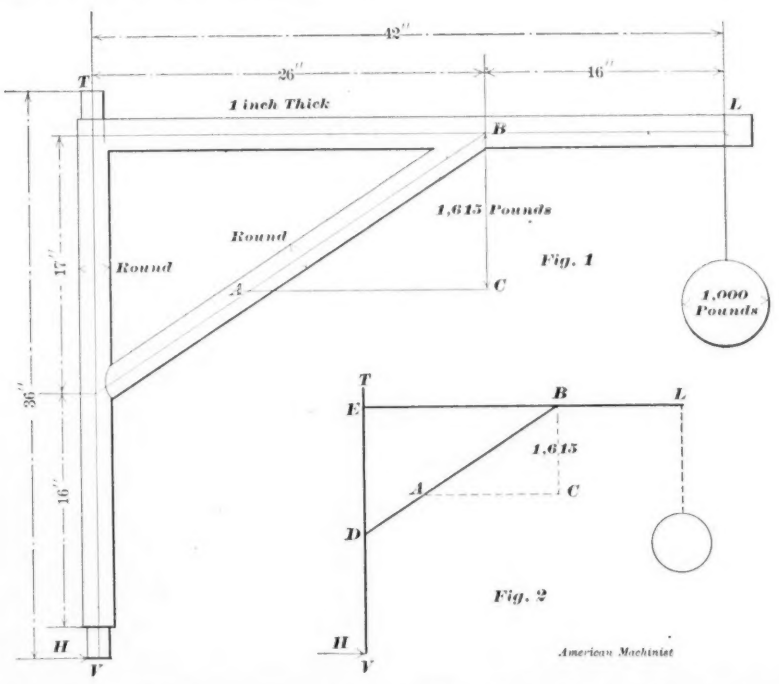

Fig. 1

Fig. 2

draw the horizontal line $A\,C$, intersecting $B\,D$ in the point A, we thus obtain the triangle $A\,B\,C$, whose side $B\,C$ represents the vertical force acting downwards at B; the side $A\,C$, the tension in the horizontal member $E\,B$, and the side $A\,B$, the compression in the strut $B\,D$. If $B\,C$ is 8½ inches long, then will $A\,C$ be 13 inches long, and $A\,B$ 15 6 inches very nearly. The forces which the sides of this triangle represent are proportional to the length of these sides. $B\,C$ we know represents 1,615 pounds, hence $A\,C$ represents $\frac{1\,615 \times 13}{8\,5} = 2{,}470$ pounds, which is the tension in $B\,E$. The side $A\,B$ represents $\frac{1{,}615 \times 15\,6}{8.5} = 2{,}905$ pounds, which is the compression in the strut $B\,D$. Let us now find the horizontal force H, which must be applied at the bottom V of the post to hold the crane in equilibrium. The load of 1,000 pounds acts with a leverage of 42 inches, hence its moment will be $1{,}000 \times 42 = 42{,}000$. Let the upper end T of the post be the origin of moments, or, in other words, the point about which the forces turn; then for equilibrium the moment of H, acting with a leverage of 36 inches, must be equal to the moment of the load, namely 42,000, hence the horizontal force H will be $\frac{42\,000}{36} = 1{,}166$ pounds. We are now in a position to determine the dimensions of the different members of the crane. The lower part $D\,V$ of the post we may consider to be a beam fixed at D and subjected to a transverse force H of 1,166 pounds at the end V. The strength of this beam, in fact, of any beam, is measured by the load it can carry, and this is represented by the formula

$$W = \frac{S \times I}{l \times c}, \qquad (1)$$

in which W = load in pounds, S = safe stress per square inch in outer fibers of the beam, for which we may take 12,000 pounds for wrought iron, I = the moment of inertia, l = length of beam in inches, and c = the distance from the neutral surface of the beam to the most remote fiber. For a circular section the moment of inertia is $\frac{3\ 1416\ d^4}{64}$, and c will be equal to ½ d, in which d = diameter in inches of the section. Substituting these letters for those in formula (1) we have

$$W = \frac{S \times 3\ 1416\ d^4}{l \times \frac{1}{2}\,d \times 64};$$

this reduces to

$$W = \frac{S \times d^3}{10\ 18 \times l} \qquad (2)$$

from which we deduce

$$\frac{W \times 10\ 18 \times l}{S} = d^3. \qquad (3)$$

Substituting in formula (3) for the letters their values we have

$$\frac{1.166 \times 10\ 18 \times 16}{12{,}000} = d^3 = 15.8,$$

and the cube root of 15 8 is 2.5 very nearly, which is the diameter of the lower end of the post required for resisting the transverse force only. The corresponding area should be sufficiently increased to resist the compressive stress due to the load and weight of the crane; but this increase will be comparatively small, an area corresponding to 2⅝ inches diameter will be sufficient for all purposes; hence the diameter of the post should be 2⅝ inches. The portion $B\,L$ of the horizontal jib may be considered as a horizontal beam fixed at B and loaded at the other end L. This jib is to be, according to sketch, 1 inch thick. The formula for finding the depth is deduced also from formula (1) by giving the proper values for I and c. The moment of inertia of rectangular sections, such as this jib is to have, is $\frac{b\,d^3}{12}$, in which b equal breadth in inches, and d = depth in inches of the beam, and for sections of this kind $c = \frac{1}{2}\,d$. Hence formula (1) will be changed to

$$W = \frac{S \times b \times d^3}{l \times 12 \times \frac{1}{2}\,d};$$

this reduces to

$$W = \frac{S \times b \times d^2}{6 \times l},$$

from which we deduce

$$\frac{W \times 6 \times l}{S \times b} = d^2. \qquad (4)$$

Substituting in formula (4) for the letters their values we have

$$\frac{1{,}000 \times 6 \times 16}{12{,}000 \times 1} = d^2 = 8,$$

the square root of 8 is 2.8 inches, which is the depth of the beam $B\,L$ to resist the load only. This depth should be increased so as to allow for the tension of 2,470 pounds in $E\,B$ of the horizontal jib; this increase of depth will, however, be comparatively small, 3 inches for the depth of jib will be sufficient. The ultimate compression C per square inch which the strut $B\,D$ can resist is found by the following formula:

$$C = \frac{f}{1 + \frac{l^2 \div r^2}{a}}, \qquad (5)$$

in which f = a coefficient depending upon the nature of the material, it is usually taken at 40,000 for wrought iron, l equal length of strut in inches, r = radius of gyration of the cross-section of the strut, the square of this radius is equal to the square of the diameter divided by 16, and for struts which are hinged at both ends, as we assume this one to be, $a = 20{,}000$. If we adopt 1½ inches for the diameter of the strut, whose length we have found to be 31 2 inches then by substituting these values for the letters in formula (5) we have for the breaking load in pounds per square inch of area of section

$$C = 1 + \frac{40.000}{\frac{973.44 \div 0.72}{20{,}000}} = 37{,}582$$

pounds. The cross-sectional area of a strut 1½ inches diameter is practically 1 square inch, the result of our computation shows that the breaking load is nearly ten times greater than the stress in the strut, hence we conclude that a strut 1½ inches diameter is strong enough and safe. Your other question we will answer in a future issue.

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Punching Sheet Steel in a Planer—A Milling Attachment to Planer.

BY JAMES F. HOBART.

An exceedingly simple method of converting an ordinary planer into a first-rate punching or shearing press, is illustrated by the line engravings herewith presented. The problem arose how to cut out a number of notches in steel disks, something as shown at *a*, Fig. 1 (page 10). The steel was $\frac{7}{100}$ of an inch thick, ordinary "saw" steel, and the edges of the notch were 4 inches long.

Estimates were secured from a press manufacturer, and it was found that a machine to notch the disks (eight notches each) would cost several hundred dollars, and the dies would cost $40 or $50 more. This estimate settled the idea of a special machine, especially as the work was experimental, and the outlay could not be warranted.

Finally, the scheme shown at *c*, *d*, *e*, was adopted, and it has proved a perfect success. Upon the platen of a planer, was bolted the die *b*, and the punch *d* was hinged to the block *e*, which was also bolted to the

planer. A roller g was secured to the beam i of the planer by means of the casting h, which is a side elevation, the front being shown at j. The steel plate a being inserted between c and d, and the planer started, the roller or wheel g comes in contact with d, and forces it down, pressing the die through the steel plate.

It will be seen that the wheel g does not come in contact with d until the wheel is directly over the line of cut, and the wheel follows the line of cut to its completion. The spring f returns the die, after a cut has been made, as fast as the wheel g is removed by the return stroke of the planer. A pin at b carries the plate, and an index rig is attached for the purpose of dividing the plate accurately.

About the right amount of shear to insure easy cutting, is caused by properly arranging the thickness of die and punch, as shown, and by properly adjusting the shifting mechanism of the planer, any desired length of time between strokes may be secured, thus, just the time needed for indexing the plate may be secured.

The die and punch, illustrated by Fig. 2, are marvels of simplicity and cheapness. It may be well to state here, that the total cost of all material used in constructing this rig, including even the lumber for patterns, was $5.05, and the entire time consumed in making the device, was less than 10 days of 10 hours each. Quite a contrast to $450 or $550! Furthermore, the inventor of the "kink," Mr. T. R. Almond, York and Washington streets, Brooklyn, assures readers of the American Machinist that the device is not patented, and that they are welcome to use it as they please. It is also at once seen that the device may be easily applied to many other operations, such as punching, bending, forming, etc.

The punch, a and b, Fig. 2, is made from a plain bit of steel one inch thick, and about four inches long. Its taper is 40 degrees, and it is held in place by two dowel pins and three machine screws, the heads being countersunk, as shown. The die c d is forged up from a flat bar, about one-half by one inch. It is held in place by four screws, which are tapped into it as shown by the engraving. This die was forged up and screwed in place, and while soft by annealing, the punch was forced into it, giving the desired shape.

After hardening, the die was found to have sprung open about $\frac{1}{16}$ inch. This insures a tight fit, after it is forced into its place. To make sure of a good temper, the die was bolted into its box, and the temper drawn while it was there by means of gas heat. Very even heating was thus insured.

Fig. 3 shows the die box. It is a piece of flat cast-iron with the lugs b b cast on; the screws to hold the die go into the holes d d d d, and the die rests upon the body of the box, or holder, at c c, a form of construction insuring great solidity. The holes a a receive bolts for securing the device to the planer platen; the stud e, shown in elevation at f, receives the index and plate-holding devices.

The hinged punch carrier, a and b, Fig. 4, are also made of cast-iron. The great width shown in plan view is for the purpose of preventing any lateral motion or spring of the punch. The various screw and dowel holes are shown. The bottom side of these castings are shown in the plan views. It will be noticed that instead of both parts of the hinge lying "on their backs," like an ordinary rule joint, which they somewhat resemble, the part d has been turned upside down. This illustrates one of the queer things that are continually happening in machine shop practice. It was originally intended to use a the other side up from what it is shown in the engraving, but, the man who was making it got the punch bolted on the wrong side, whereupon it was determined to use it as put together. The mistake proved an improvement. It gave about the right shear to the punch without blocking up either c or e (Fig. 1), as was intended. It also leaves a nice place for the spring f, and, better yet, it leaves the top of the machine perfectly smooth, and free from anything likely to catch the roller g.

Upon the same planer I noticed the device illustrated by Fig. 5. It was bolted to the hand feed arrangement carried by the beam, the ordinary tool box having been removed, and the tool illustrated put in its place. It is driven by means of an overhead drum, having a length equal to width of the planer. By means of suitable feed arrangements attached to table movement, and beam screw, the end mill shown at a can be made to chase irregular pieces of metal all over the planer table, and does good work at almost any kind of surface work that can be done with an end mill.

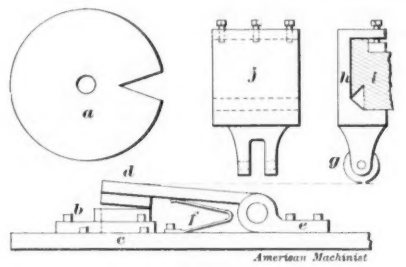

Fig. 1

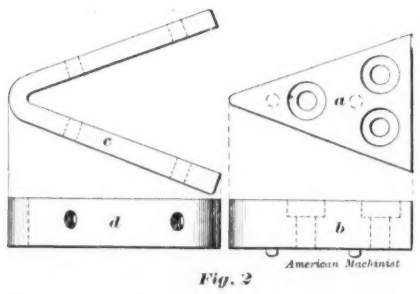

Fig. 2

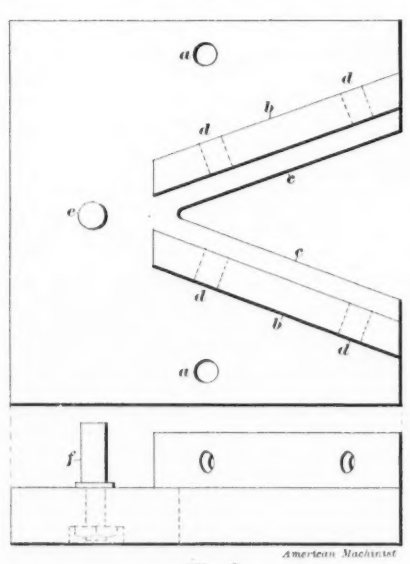

Fig. 3

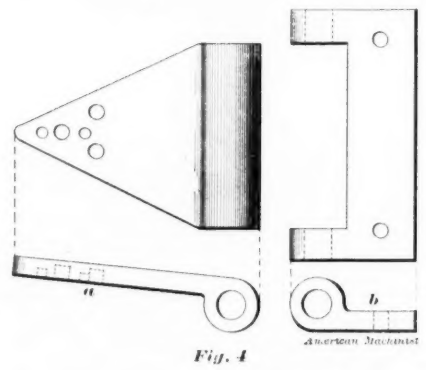

Fig. 4

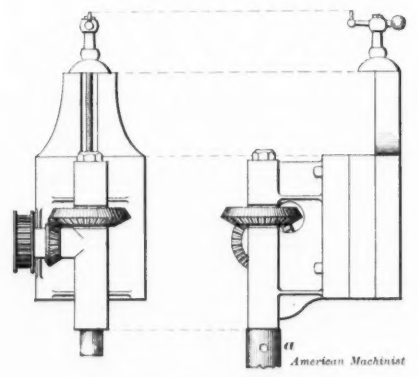

Fig. 5

Punching Sheet Steel on a Planer.

Compulsory Arbitration of Labor Disputes.

Carroll D. Wright, U. S. Commissioner of Labor, delivered an address in Chicago, recently, in which he clearly exposed the fallacy of compulsory arbitration. Mr. Wright said that no way had yet been devised to prevent strikes. The most that could be done is to reduce them in number and severity. Voluntary arbitration seemed to him a logical conclusion.

Passing to the question of compulsory arbitration, he said: "The advocates of this say that both parties to a dispute must be compelled to come into court; and that when the court issues its decree it must be enforced as any other decree would be enforced. Let us see how this will work. A. is a manufacturer who pays his men on an average of $2 a day. Owing to some cause he finds he cannot pay this any longer and reduces wages to $1.80. The men oppose this. They say they cannot live on less than $2. The employer says he cannot pay more. The matter is taken up before a compulsory court of arbitration. Suppose the court says that the men must take the $1 80. How can this be enforced? Say there are 50,000 employes. Who can make them work? The constables, the sheriff, the posse comitatus? The military forces may all be called out, but none can make them work. The decree is dead from the beginning. Suppose the court says the manufacturer must pay the $2, but if he really cannot afford to do so what must happen? The employer must do one of two things. Either he must adulterate his goods to make up the difference, or he must combine with other employers in a trust to keep up the prices. The law will have compelled one of these things.

"There is also an economic question to be considered. If the State says the employer must pay a fixed price for work, it has practically fixed the price of the commodity. The law can fix the price, but it cannot compel the consumers to buy at that price. The next logical step is to make it a penal offense not to buy at the price. Then why not let the law fix the price of everything directly? It would be cheaper to do this directly than to do it by compulsory arbitration. Compulsory arbitration will establish the process at which the employe can sell the only commodity he possesses—his labor. He must accept $1.80 or not work at all if the court tells him to do so, even if the actual cost of living is $2.

"On whatever side we look at it compulsory arbitration means the death of industry and the enslavement of labor. There may be some modification of the general principle developed in the future, but I do not know it now. There should be a protest by the community against it. Few labor organizations endorse it. A few scholars and thinkers are almost the only ones who really endorse it. To accept it would be to go back to the ages from which we have been divorced. It would be worse barbarism than we have known for ages."

American Locomotives in Brazil.

A railroad train remarkable in many respects was run from the Brooks Locomotive Works, Dunkirk, N. Y., on the 9th inst. over the Erie Railroad to Jersey City. The entire load of the train consisted of locomotives dismantled and packed for shipment to Brazil, where they will be operated on the Brazilian government roads. There were 25 locomotives carried on the train which consisted of 28 cars. Other locomotives have already gone by previous steamers, and others will be shipped as rapidly as possible until sixty, the number called for by the contract, have been sent. This, so far as we know, is the first full train load of American machinery ever sent into a foreign country at a single shipment. Mr. Lewis Gleason, an engineer connected with the Brooks establishment, called at this office a few days ago just previous to his sailing for Brazil, where he will superintend the erection of the engines, and remain there to see that everything is satisfactory.

Plates of Uniform Strength.—III.

By Albert E. Guy.

SECTION 6.

An annular plate of inner radius r is encastré or fixed at its outer circumference of radius R, supports a load p per unit of area, and a load P distributed along its inner circumference (Fig. 8).

The moment due to load P is

$$M_1 = P\,l$$

The total load uniformly distributed is

(49) $p\,(\pi\,(l-r)^2 - r^2) = p\,\pi\,(l^2 + 2\,r\,l)$.

The bending moment caused by this load

(50) $M_2 = p\,\pi\,(l^2 + 2\,r\,l)\,(l - x) = p\,\pi\,l^2\,(l + 2\,r)\,(1 - n)$.

The value of n in this case is the same as that found in the preceding paragraph.

The total moment $M = M_1 + M_2$.

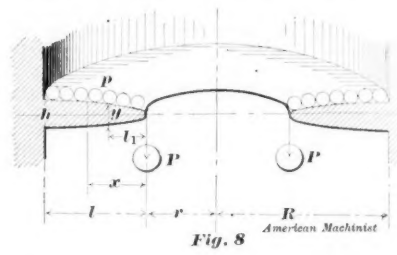

Fig. 8

(51) $$M = P\,l + p\,\pi\,l^2\,(l + 2\,r)\,(1 - n) = f\,\frac{b\,h^2}{6}.$$

When $b = 2\,\pi\,R$,

$$\text{then } M = f\,\frac{\pi\,R\,h^2}{3},$$

and corresponding thickness

(52) $$h = \pm\sqrt{\frac{3\,P\,l + 3\,p\,\pi\,l^2\,(l + 2\,r)\,(1 - n)}{f\,\pi\,R}}.$$

The thickness at any circumference $2\,\pi\,(r + l_1)$ is

(53) $$h_1 = y = \pm\sqrt{\frac{3\,P\,l_1 + 3\,p\,\pi\,l_1^2\,(l_1 + 2\,r)\,(1 - n)}{f\,\pi\,(r + l_1)}}.$$

But if $r = 0$, equation (52) becomes:

$$h = \pm\sqrt{\frac{3\,P\,l + 3\,p\,\pi\,l^2\,(1 - n)\,l}{f\,\pi\,R}}.$$

Then $l = R$, and $(1 - n) = 1 - 0\,7071 = 0.2929$.

Substituting, we obtain

$$h = \pm\sqrt{\frac{3\,P}{f\,\pi} + \frac{0.8787\,p\,R^2}{f}} = \pm\sqrt{\frac{0\,955\,P + 0.8787\,p\,R^2}{f}}.$$

This equation is the same as we have found in § 3 (26).

In calculating internal flanges it occurs most commonly that they are loaded in a manner similar to that of the plate treated in this chapter. But the load P, which may be produced by the pressure or tension of the bolts binding the cover of the cylinder, for instance, to its flange, is not always applied at the free end of the plate. Generally the place of application of this load is about at the center of l. It will be easy to modify accordingly the necessary equations by replacing in them the value of $P\,l$ by the correct one.

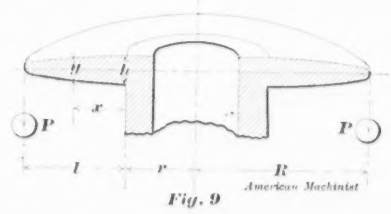

Fig. 9

(54) The stress

$$f = \frac{3\,P\,l + 3\,p\,\pi\,l^2\,(l + 2\,r)\,(1 - n)}{h^2\,\pi\,R}$$

(55) And the load

$$P = \frac{h^2\,f\,\pi\,R}{3\,l + 3\,p\,\pi\,l^2\,(l + 2\,r)\,(1 - n)}$$

SECTION 7.

An annular plate of outer circumference $e\,\pi\,R$ is encastré or fixed along its inner cir-

cumference of radius r and holds along its free end a load P (Fig. 9).

As before, we have $R - r = l$.

The moment due to force or load P is

$$P\,l = f\,\frac{b\,h^2}{6},$$

but $b = 2\,\pi\,r$;

(56) hence $P\,l = f\,\frac{\pi\,h^2\,r}{3}$.

The thickness at circumference $2\,\pi\,r$ is

(57) $$h = \pm\sqrt{\frac{3\,P\,l}{\pi\,f\,r}}$$

$$= \pm\,0.977\sqrt{\frac{P\,l}{f\,r}}$$

For any radius $r + x$ $b = 2\,\pi\,(r + x)$.

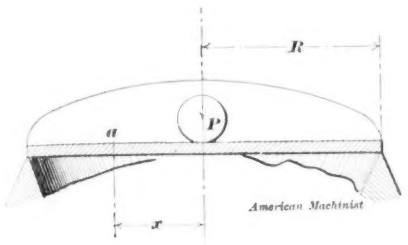

Fig. 10

(58) The thickness then is

$$h_1 = y = \pm\,0.977\sqrt{\frac{P}{f}\left(\frac{l-x}{r+x}\right)}.$$

(59) The stress $f = \frac{0.954\,P\,l}{h^2\,r}$

$$= \frac{0.954\,P}{y^2}\cdot\left(\frac{l-x}{r+x}\right)$$

(60) The load $P = \frac{f\,h^2\,r}{0.954\,l}$

$$= \frac{f\,y^2}{0.954}\cdot\left(\frac{r+x}{l-x}\right)$$

SECTION 8.

A plate of radius R is supported, not fixed along its circumference, and is loaded at center with a weight P (Fig. 10).

This case is a deduction of the preceding one, for if we make $r = 0$, the plate will then be full, supported at center and loaded at its circumference; but we find that the formulæ used in the first case do not apply in this one, if we place the weight at the exact center.

It may be stated here that no formulæ are in use for determining the dimensions of a plate loaded in such a manner. Professor Grashof in his "Theory of Elasticity" has established rules intended to solve this problem, but they will be found erroneous in the case of large plates.

The problem as expressed by Professor Grashof reads thus:

A circular plate of radius R and thickness h is supported at the edge and loaded with a concentrated load P applied at a circumference the radius of which is r.

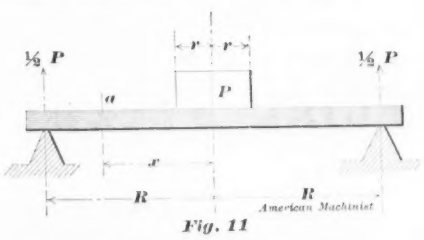

Fig. 11

(61) The greatest stress is:

$$f = \left(\frac{4}{3}\text{ log. nat. }\frac{R}{r} + 1\right)\frac{P}{\pi\,h^2}.$$

And when $\frac{R}{r}$ =	10	20	30	40	50
$\left(\frac{4}{3}\text{ log. nat. }\frac{R}{r} + 1\right)$ =	4.07	5.00	5.53	5.92	6.22

Suppose that the thickness h is required; assuming a value for P and f, then,

$$h^2 = \left(\frac{4}{3}\text{ log. nat. }\frac{R}{r} + 1\right)\frac{P}{\pi\,f}$$

But $\frac{P}{\pi\,f}$ = constant = c,

(62) and $h = \pm$

$$\sqrt{\left(\frac{4}{3}\text{ log nat. }\frac{R}{r} + 1\right)c}.$$

In other words the thickness h varies—not with the load—but according to the degree of concentration of the load. It follows that if the load is applied at the very center (which in theory is perfectly feasible and permissible, leaving aside all considerations about the shearing stress), $\frac{R}{r}$ becomes $\frac{R}{0} = \infty$; and $h = \infty$, independently of the load, stress and radius of the plate.

We may remark here, that a load P was applied exactly at the center, theoretically, of a plate encastré or fixed at its circumference of radius R (paragraph 1) and that formulæ have been established for determining the thickness of the plate corresponding to a stress f due to the load P, and *vice versa*.

Fig 11 represents a prismatic beam resting upon two supports, and loaded at its center with a weight P.

The reaction of each support $= \frac{P}{2}$.

At a point a the bending moment is $\frac{P}{2}\,(R - x)$; and at the center of the beam it becomes maximum and $= \frac{P}{2} \times R$, when the load is distributed on a distance $2\,r$. Whether we make $r = x$ or $r = 0$, the bending moment at a will always be constant if the load P remains constant.

In a diametral section a circular plate presents a certain analogy with this prismatic beam. Reverting to Fig. 10, the bending moment at a point a distant from the center

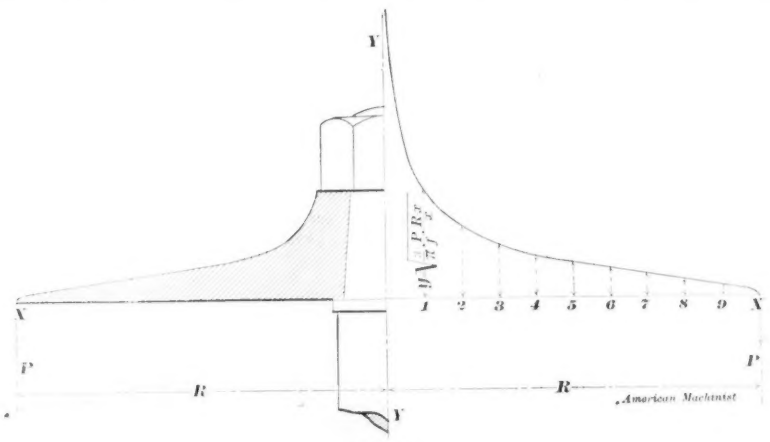

Fig. 12

a length x, is $P\,(R - x)$, and, it seems, will remain constant independently of the mode of arrangement of the load P, the space occupied by this latter varying from 0 to $2\,x$.

If we attempt to calculate the strength and dimensions of the plate in a similar manner to that used for the prismatic beam, a serious objection presents itself, the analysis of which will furnish us the key to the solution of the present problem.

The point of application of the resultant of the forces acting upon the beam is the section at the center. In other words, the maximum bending moment

(63) $$M_{max.} = \frac{P}{2} \times R.$$

The molecular stresses developed in the beam form, at the center, a couple, whose moment is equal and opposite to the bending moment. If the beam is rectangular in section, then

(64) $$\frac{P}{2} \times R = f\,\frac{b\,h^2}{6}.$$

In the case of the circular plate, $b = 2\,\pi\,x$ for any radius x, and the equation expressing the thickness is:

(65) $$h = \pm\sqrt{\frac{6\,P\,(R-x)}{f \times 2\,\pi\,x}} = \pm\sqrt{\frac{3}{\pi}\cdot\frac{P}{f}\cdot\frac{(R-x)}{x}}.$$

If the load is applied at center: $\frac{R - x}{x} = \infty$, for then $x = 0$.

The thickness h varies along the radius from 0 at the point of support to infinity at the center. The curve representing this variation is drawn in Fig. 12.

It follows that at center the section $= 2\,\pi\,x\,h$ and as $x = 0$, $2\,\pi\,x\,h = 0$.

But since the section at center $= 0$, and since the plate must not only hold itself together, but also support a given load, we conclude that:

In a circular plate, supported at its circumference, and loaded at center with a weight P, the point of application of the resultant of the forces acting upon it (excluding the shearing stresses) is not the center of the plate.

If a plate of radius R is placed upon a concentric support of radius r (Fig. 13), and is loaded at the edge with a weight P, and at the center with an equal weight φ, the bending moment $P\,l = f\,\frac{\pi}{3}\,r\,h^2$, and the thickness at the point of support is

(66) $$h = \pm\sqrt{\frac{3}{\pi}\cdot\frac{P}{f}\cdot\frac{l}{r}}$$

The bending moment $\varphi\,r = f\,\frac{\pi}{3}\,r\,h_1^2$, and the thickness $h_1 = \pm\sqrt{\frac{3}{\pi}\cdot\frac{\varphi}{f}}$, is uniform along radius r. The two moments are equal when $l = r$; then

(67) $$h = h_1 = \pm\sqrt{\frac{3}{\pi}\cdot\frac{\varphi}{f}} = \pm\sqrt{\frac{3}{\pi}\cdot\frac{P}{f}}.$$

The bending moments are maximum along circumference $2\,\pi\,r$, and they produce along $a\,b$ opposite and equal forces c and c, above the axis of the plate, straining the material in tension, and forces d and d, below the axis, straining the material in compression.

If now the force or weight P be made to act in a direction opposite to φ (Fig. 14),

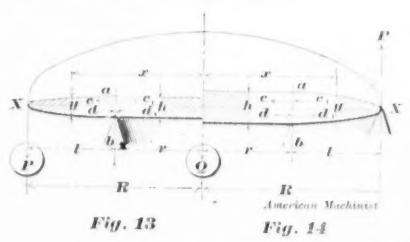

Fig. 13 *Fig. 14*

the bending moments will still be equal, and will produce along $a\,b$ forces equal in intensity as in Fig. 13. But as these forces have similar directions, their resultant is equal to their sum. Consequently the bending moment about $2\,\pi\,r$ is equal to the sum of $P\,l$ and $\varphi\,r$, and we have for the thickness at $a\,b$,

(68) $$h = \pm\sqrt{\frac{3}{\pi}\left(\frac{P}{f} + \frac{\varphi}{f}\right)}$$

$$= \pm\sqrt{\frac{6}{\pi}\cdot\frac{P}{f}}$$

This thickness will be uniform along radius r (see § 1), but along l, $h_1 = y$ will be

(69) $$y = \pm\sqrt{\frac{6}{\pi}\cdot\frac{P}{f}\cdot\left(\frac{R-x}{x}\right)}$$ at any

circumference $2\,\pi\,x$. When $x = R$, then $y = 0$, for the bending moment $= 0$.

The shearing stress at any circumference $2\,\pi\,x$ is expressed by the equation

(70) $$f_1 = \frac{P}{2\,\pi\,x\,y}.$$

The force P acting in an opposite direction to force φ, it is just as if the support of radius r was removed and the plate resting upon a support of radius R.

It is evident from the foregoing that the circumference of radius $r = \frac{R}{2}$ is the point of application of the resultant of the forces acting upon the beam, but other circumferences could be made to answer the same purpose, and this is evident in the case of full plates supported at the edge. Effectively there is a superfluity of material in a full plate, and the portion l of the radius is thicker than required, and consequently stronger than the rest of the plate.

A full plate is also stronger than a plate of uniform strength for the reason that the portion l of the plate is more rigid in the first than in the second case, and the plate of radius r may to a certain extent be considered as fixed or encastré at its circumference, thereby requiring a lesser thickness.

In using equations (68) and (69), found by this last method, to determine the thickness of the plate, the disposition of the load affects the bending moment. If the load is applied upon a circle of radius r_0, the total moment will be

$$M = P\,(R - r_0).$$

The maximum value of h will be

$$h_{max.} = \pm\sqrt{\frac{3}{\pi}\cdot\frac{P}{f}\cdot\frac{2\,(R-r_0)}{R}}$$

$$= \pm\,1.38\sqrt{\frac{P}{f}\cdot\left(\frac{R-r_0}{R}\right)}.$$

Careful experiments have been conducted in Germany, by Professor C. Bach, and published in the *Zeitschrift des Vereines Deutscher Ingenieure* in October, 1890, Nos. 41, 42, 43, 44. Circular plates, each of uniform thickness, supported near the edge, were subjected at the center to the pressure of a hydraulic ram. Deflections corresponding to different loads, including breaking, were recorded. Then Professor Bach imagined some formulæ to calculate the strength of the plates. Care had been taken to test pieces of each plate in order to find the coefficient of resistance of the material. This value introduced in the formulæ gave for the thickness of the plate:

$$h = \pm\sqrt{\frac{1}{\varphi}\cdot\frac{3}{\pi}\cdot\frac{P}{f}}.$$

$\frac{1}{\varphi}$ is a constant which the professor found necessary to use in his equation to account for all the stresses he could not calculate by his method. P and f being known, the experiments on steel, cast iron and brass plates of a uniform diameter of 56 centimeters, or about 22 inches, gave the average value of $\frac{1}{\varphi} = 1.67$.

Then $h = \pm\,1.26\sqrt{\frac{P}{f}}$.

The hydraulic ram was rounded at the end, and the contact space was found (when the plates broke) to be three centimeters in diameter, that is, a little less than $1\frac{3}{16}$ inches. If we now use Grashof formula (61) to calculate h we have,

$$h = \pm\sqrt{\left(\frac{4}{3}\text{ log. nat. }\frac{56}{3} + 1\right)\frac{P}{\pi\,f}}$$

$$\frac{56}{3} = \frac{D}{d} = \frac{R}{r} = 18.66.$$

$$\frac{4}{3}\text{ log. nat. } 18.66 + 1 = 4.9,$$

and $h = \pm\,1.249\sqrt{\frac{P}{f}}$.

Comparing the three methods we have for the value of h,

Professor Grashof:

$$h = \pm\,1.249\sqrt{\frac{P}{f}}.$$

Prof. C. Bach:

$$h = \pm\,1.262\sqrt{\frac{P}{f}}.$$

formula ( ):

$$h_{max.} = \pm\,1.38\sqrt{\frac{P}{f}\cdot\left(\frac{28 - 1.5}{28}\right)}$$

$$= \pm\,1.34\sqrt{\frac{P}{f}}.$$

Remarks.—1°.—Fig. 12 represents, at the left, the radial section of the web of a piston, the thickness of which may be calculated from the formulæ established in paragraph (7). At the right is drawn the curve showing the variation of the thickness along the radius. The bending moment due to a load P, acting at the edge, is equal to

$$M = P\,(R - r),$$

r is the radius of the hub.

2°. This piston could be calculated by formula (69) with equal correctness, but

would, of course, be made of a different shape, such as shown in Fig. 14.

3. Our method permits us to apply the load P exactly at the center of the plate, theoretically, requiring to do so but a slight increase of the thickness of the plate due to the augmentation of the bending moment.

4. This method will also allow us to find the dimensions and strength of annular plates supported at one edge and loaded at the other. These plates are not used much in practice, but their study will help us to calculate the full plates of uniform thickness and the compound plates.

We have received from the Franklin Institute, Philadelphia, Pa., an announcement and programme of lectures for the season of 1894–5. It is evident from this that those who enjoy membership in this society will be unusually favored during the coming winter in the way of useful and entertaining lectures, besides the other advantages of membership.

Bement, Miles & Co., of Philadelphia, following the example of other prominent tool builders who have established offices in New York, have opened an office at 39 Cortlandt street. Mr. E. H. Mumford, recently with Henry R. Worthington, is in charge, and will be prepared to give full information regarding Bement, Miles & Co.'s productions.

A technical and trade school endowed by the late James Lick, of California, and to be called The Lick School, will be opened for the reception of students at San Francisco, July 7, 1895. The principal, George A. Merrill, is now in the East buying tools and equipment. Tuition will be free in this school, the students paying only for books, drawing instruments, and cost of materials used.

Naval ordnance officers are said to be experimenting with a rifle which employs what may be called the compound principle, there being two charges of powder, one of which is not ignited until after the shot has passed part of the way through the tube. It is expected that much higher initial velocities will be attained without excessive pressure at the breech.

What is said to be the fastest trip ever made through Long Island Sound from New York to New London, Connecticut, was run by the new Norwich Line steamer "City of Lowell," the evening of the 9th inst. She covered the distance, 122 miles, in 5 hours 34 minutes, the usual time being 7 hours. The vessel was built at the Bath Iron Works, is a twin screw propeller 336 feet long, and with 5,400 horse-power engines. Illustrations and dimensions of the engines and the steamer were given in our issue of June 21st, current volume.

M. Desdouits, engineer in charge of locomotives and rolling stock of the State Railway of France, in a recent paper in the *Revue Generale des Chemins de Fer*, compares the economical efficiency of locomotives with stationary and marine engines, his conclusion being that locomotives are as economical as other steam engines of similar types. In regard to the high pressure used in compound locomotives, M. Desdouits claims that such high pressure cannot be economically used in simple locomotives, and that it is, therefore, perfectly fair for compound engine builders to use the higher pressure in every test, this high pressure being an essential and distinctive part of the compound system.

A train parted in the middle, and the signal rope snapped off like a thread, the end of it striking an old lady on the bonnet. "What is the matter?" she exclaimed. "Oh, the train is broke in two," replied a gentleman, who sat in the next seat. "I should say so," the old lady said, looking at the broken cord. "Did they s'pose a little bit of string like that would hold the cars together?"—*Transportation.*

Denver, Col., is said to have the longest street railway cable in the world. The rope which acts as the motor for the Welton street line is 36,900 feet, or about seven miles in length, being just 41 feet longer than the longest cable on the Broadway Road in New York.

The Carborundem Co., of Monongahela City, Pa., have contracted with the Niagara Power Co. for 1,000 electric horse power, and have secured a piece of ground 450 by 400 on which they will erect their buildings. They will break ground at once, and will start the new plant as soon as possible.

MANUFACTURES

The general locomotive repair shops of the Fitchburg road are to be removed from Charlestown to Keene, N. H.

The Davis & Furber Machine Co., Lawrence, Mass., have increased their working time to 54 hours per week.

A go'd medal has been awarded to the Cleveland Twist Drill Co., Cleveland, O., for their exhibit of twist drills at the Antwerp Exhibition.

The J. A. Fay & Egan Co. inform us that they have been awarded a gold medal for excellence of their display of wood-working machinery at the Antwerp Exhibition.

J. H. Dunbar, Youngstown, O., writes: I take pleasure in saying that I have just received my fourth order for American packing rings from the N. Y., O. & W. Ry., making a total of 194 rings to this road.

The Springfield File Company, who have done business at 48 Taylor street, Springfield, Mass., have sold out to parties in Worcester. They will make files in a new way, and for this purpose machinery is being made in Worcester.

From the Lunkenheimer Co., Cincinnati, Ohio, we have received a small illustrated pamphlet giving clear descriptions of their specialties in the way of valves for various purposes, whistles, lubricators, etc. It will be found of interest to those connected with steam plants, and will be sent upon application.

We have received from the W. E. Caldwell Co., Louisville, Ky., a catalogue of tanks of various kinds to be used for water stations and by manufacturing concerns. The catalogue gives full information regarding construction, capacity, etc., of such tanks, with tables of weights, cost of iron work, estimated costs of foundation, etc.

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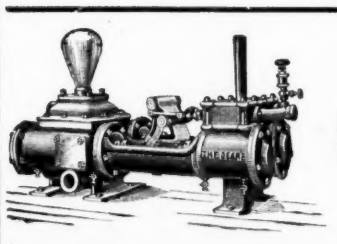

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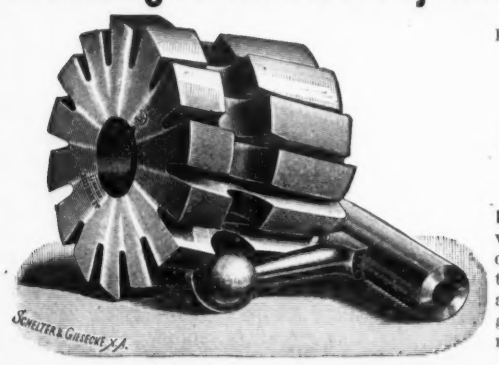

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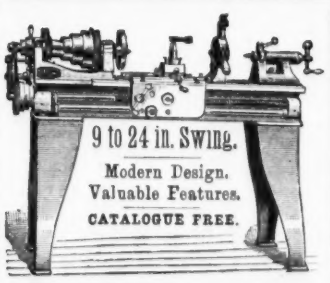

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Official notice has been received by the Penberthy Injector Co., Detroit, Mich., that they have received an award from the World's Columbian Exposition on injectors, the language of the award being as follows: "They are efficient and reliable, simple in design and construction, thoroughly automatic under varying conditions, showing great lifting power through hot or cold suction pipe, and are easily operated and capable of working under a broad range of possibilities."

From the Shelby Steel Tube Co., Shelby, Ohio, we have received a pamphlet and table giving dimensions of their cold drawn seamless steel tubes, which are used for boilers, hydraulic presses, hollow spindles, etc. The table is conveniently arranged and gives the exterior diameter of the tubes in inches, the corresponding number of the English standard wire gauge, and the equivalent in decimals of an inch. The company state that they use micrometer calipers entirely, but when referring to a gauge the English standard is used.

The Ferracute Machine Co., of Bridgeton, N. J., write us that they are quite busy and are building several unusually large presses for various purposes, among them being a heavy double column press for electrical work and another for embossing ceiling plates, a large drawing press for gasoline stove work, and several smaller punching and cutting presses for general work in tinware, hardware, etc. They have just shipped a press with double feed attachment to the Frankford Arsenal, Bridesburg, Philadelphia, for cartridge shells, etc.

A press dispatch from Pittsburgh says: Powerful competition to the steel rail trust is coming from the Mahoning Valley. At Youngstown, O., an immense steel plant has been erected by the Ohio Steel Company at a cost of $1,500,000. In January next it will be placed in operation day and night with a carefully selected crew of experienced workmen. The plant will have a capacity of 1,000 tons of steel rails per day, and by reason of the many modern improvements employed, it will do the work at a great saving in cost over present methods. The new development threatens destruction to the colossal pool that now virtually controls steel rail and billet production in the United States, and it seems certain that steel rails will be cheaper in a few months than ever before in America.

The E. W. Bliss Co., Brooklyn, N. Y., write us: Our European business has been very large, as we have shipped to Switzerland within the last two months a special watchmaker's drop hammer and several punching presses fitted with sub-presses for watch work, also a No. 1½ toggle drawing press. A large shipment to Germany, to one of the largest clock-making concerns in the world, of tools and machinery for the manufacture of clocks and their cases. France has also received a large shipment of tools for the making of granite enameled ware and kitchen utensils. Several watch factories have also been supplied with tools from us. Austria has received a No. 1½ and 3½ toggle drawing press; a No. 18, 19, 20 and 21 adjustable power presses, and a No. 38½ and 39 power presses with a number of dies, and a No. 161 double action press with dies and special feed for making primers. A large improved perforating, automatic press is now nearly completed and will be shipped to England for the manufacture of perforated metals up to 50 inches in width. We think that this speaks well for American tools in competition with those of foreign make, and we are greatly encouraged in this direction.

The Tradesman, Chattanooga, Tenn., in its report on the industrial condition of the South for the week ending October 8th, says: Each week brings more satisfactory information as to the condition of Southern manufacturers. The number reported for this week is larger than for three months past, and the number of changes and improvements is also large. Textile mills are doing well; some of them report that prices are low, but none that they cannot work at a profit. The iron output continues large and is steadily absorbed by the trade. Lumbermen are in better shape than during the spring and summer, but there is still room for improvement. Crops are large and have been made on a cash basis. *The Tradesman* reports fifty-one new industries as established or incorporated during the week, among which are: The American Smelting Co., of Wheeling, W. Va., capital $1,000,000; the Real Estate Mining and Manufacturing Co., of Rome, Ga., capital $200,000; the Lombard Iron Works and Supply Co., of Augusta, Ga., capital $100,000, and the rebuilding of the Tanners' and Dyers' Extract Co., of Charleston, W. Va., at a cost of $100,000. The Coketon Lumber Co., capital $100,000, has been chartered at Coketon, W. Va.; the Sherwood Lime and Lumber Co., with the same capital, at Sherwood, Tenn.; the Santa Fe Coal and Manufacturing Co., capital $60,000, at Rockdale, Texas, and machine shops to cost $50,000 are to be built at Atlanta, Ga. A $30,000 curtain pole factory will be built at Staunton, Va.; a $25,000 tobacco factory at Calhoun, La.; a $20,000 orange box factory at Windermere, Fla.; a $10,000 coal mining company has been chartered at Winona, W. Va., and an $8,000 ice factory at De Land, Fla. *The Tradesman* also reports an agricultural implements factory at Dayton, Tenn.; canning factories at Memphis, Tenn., Greenville, Tex., and Wellsburg, W. Va.; electrical plants at Greenville, Miss., and Americus, Ga., and flour and grist mills at Edmonton, Ky., Henry and Winston, N. C., Belbuckle, Tenn., Berryville and Houston, Va. Cotton mills are reported at Henrietta, N. C., Rock Hill, S. C., and Waco, Tex.; a cotton batting

LECOUNT'S NEW EXPANDING MANDREL.

Amateur's Size.
Taking anything from ⅛ to 1 inch inclusive.
Price.....$5.00
Patented Dec. 25, 1877.

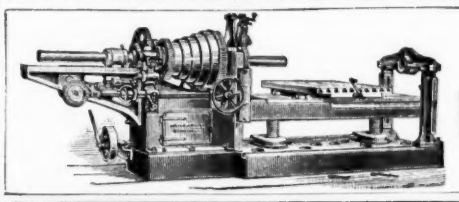
Machinist's Size.

No.		Price.
1	⅛ to 1 in.	$10
2	1 " 1½ "	14
3	1½ " 2 "	18
4 (with screws)	2 " 3 "	32
5 " "	3 " 4 "	44

IF YOU WILL TRY THIS TOOL YOU WILL NOT REGRET THE EXPENSE.
C. W. LECOUNT, SOUTH NORWALK, CONN.
SPECIAL MANDRELS FOR SPECIAL JOBS MADE TO ORDER.
These goods are for sale by CHAS. CHURCHILL & CO., L't'd. 21 Cross St., London, England.

DON'T ACCEPT ANY SUBSTITUTE FILE.
INSIST ON HAVING
NICHOLSON.

NICHOLSON FILE CO.
PROVIDENCE, R. I., U. S. A.
3000
VARIETIES FILES
[X. F.] & INCREMENT CUT FILES.

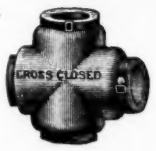


BETTS MACHINE CO.,
WILMINGTON, DEL.
MACHINE TOOL BUILDERS.
Improved Patterns! Late Designs!

DID YOU EVER?
Hear of a Coal-dealer recommending an INDICATOR? Send for our Catalogue of the ROBERTSON-THOMPSON & "IMPROVED STRAIGHT-LINE" INDICATORS. Reducing wheels, Planimeters, Hine Elimination oil filters & extractors, shaving grate bars, feed water heaters and a number of other specialties, to
Hine & Robertson Co. 61 CORTLANDT ST. N.Y.

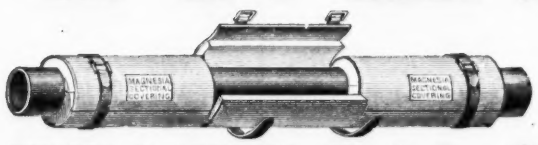
JESSOP'S STEEL
FOR TOOLS, DRILLS, DIES, &c.
ALL KINDS IN STOCK.
Manufactory, SHEFFIELD, ENG.
Chief Am. Office, 91 JOHN ST., N. Y.
WM. JESSOP & SONS, LTD.
Established a century ago.
Medal World's Columbian Exposition 1893.

"Star" Foot Lathe Swings 9x25 in.
LA LATHE THE
Screw Cutting Automatic Cross Feed, etc.
Scroll Saws. Circular Saws. Lathes. Mortisers.
Catalogue Free of all our Machinery.
Seneca Falls Mfg. Co., 687 Water St., Seneca Falls, N. Y.



MAGNESIA

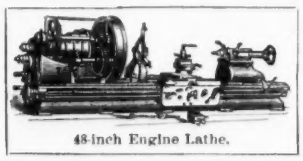
Sectional Steam Pipe and Boiler Coverings.
ABSOLUTELY FIRE-PROOF.



THE GREAT COAL SAVER
Manufactured by THE KEASBEY & MATTISON CO.
AMBLER, PA.

SELLING AGENTS.
New York, Robert A. Keasbey, 54 Warren St.
Boston, S.C. Nightingale & Childs, 134 Pearl St.
Philadelphia, Macan & Co., 1420 Callowhill St.
Baltimore, Wallace & Bro., 432 E. Pratt St.
Washington, Wm. B. Morgan, Builders' Ex'e.
New Orleans, Delbert Engineering Co.
Memphis, Symmes & Co., 192 Front St.
CINCINNATI: 114 West Second Street.
CLEVELAND: 117 Water Street.
SELLING AGENTS.
Milwaukee, F. Sprinkman, 133 Sycamore St.
St. Louis, F. Boecler, 108 Walnut St.
Detroit, S. P. Conkling, 20 Atwater St., East.
Denver, C W. Badgley & Co., 18th & Market Sts.
Salt Lake City, Utah & Montana Mach'y Co.
Butte City, Mont., R. W. James.
S. F'ncisco, DeSolla & Deussing, 3 California St

FITCHBURG MACHINE WORKS,
MANUFACTURERS OF THE CELEBRATED
FITCHBURG MODERN ENGINE LATHE
AND OTHER METAL-WORKING MACHINES
SEND FOR CATALOGUE E.


48-inch Engine Lathe.

FITCHBURG, MASS.

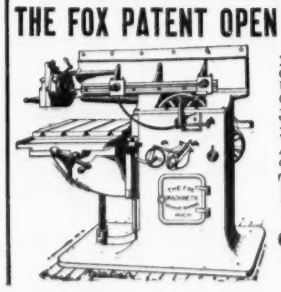
BUILDERS IRON FOUNDRY,
(Patented.)
PROVIDENCE, R. I.

GLOBE SPECIAL CASTINGS FOR WATER-WORKS, MILLS AND RAILROADS.
"I think they are the most practical special in use at the present time."
All common sizes in stock. Catalogue mailed on application.
M. J. DRUMMOND, SALES AGENT, 192 BROADWAY, NEW YORK CITY.

MACHINERY FOR SALE.
Planer, 28 in. x 24 in. x 7 ft., new.
Drill Press, 40 in. swing, new.
Engine Lathe, 24 in. x 25 ft. bed, 2d hand.
Roots' Blowers, Nos. 1, 2, 5 and 6, "
Haskin Vertical Engine, 9x9, "
Baxter Engine and Boiler, 6 & 8 H. P. "
Vertical Boiler, 40 H. P., "
Open-die Bolt Cutter, ¼ to 1¼, "
Write us before buying.
COOKE & CO.,
Machinery and Supplies,
163 & 165 WASHINGTON ST.,
NEW YORK.

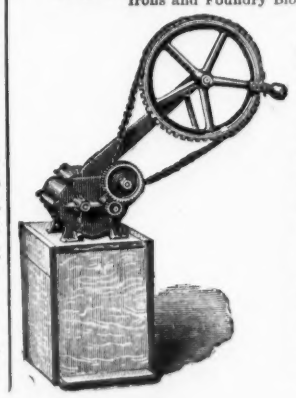
HENRY CAREY BAIRD & CO.,
INDUSTRIAL PUBLISHERS, BOOKSELLERS AND IMPORTERS,
810 Walnut St., Philadelphia.
☞ Our New and Revised Catalogue of Practical and Scientific Books, 88 pages 8vo., and our other Catalogues and Circulars, the whole covering every branch of Science applied to the Arts, sent free and free of postage to any one in any part of the world who will furnish his address.

THE FOX PATENT OPEN SIDE SHAPER.



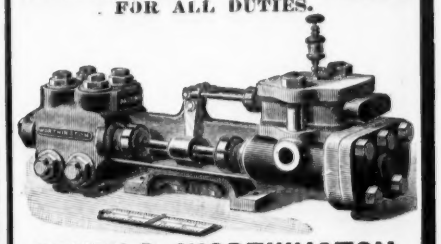
No springing of ram. No overhanging table. Quick Return, stroke Under Perfect Control. Can be Instantly Adjusted 1 16 in. or 24 in. No Screws to Monkey with. Powerful. Quick, Accurate. Write for circular.
The Fox Machine Co.,
325 N. Front St.,
Grand Rapids, Mich.
135 Finsbury Pavement, London, England.

ROOTS' NEW ACME HAND BLOWER.
For Blacksmiths, etc. Slow Speed, Positive Blast. Is Durable, Compact and Cheap; also Portable Forges, Tuyere Irons and Foundry Blowers.



P. H. & F. M. ROOTS, Mfrs.
CONNERSVILLE, IND
Chicago Office: 1405-10 MANHATTAN BUILDING.
S. S. TOWNSEND, Gen. Agt. } 163 & 165 Washington St., NEW YORK.
COOKE & CO., Selling Agts }
Please Mention This Paper.

WORTHINGTON STEAM PUMPS
FOR ALL DUTIES.



HENRY R. WORTHINGTON,
NEW YORK, 86 and 88 Liberty St.
BOSTON, 70 Kilby St.
PHILADELPHIA, 607 Arch St.
CLEVELAND, 24 South Water St.
CHICAGO, 185 to 189 Van Buren St.
ST. LOUIS, Eighth and St. Charles Sts.

mill at Charlotte, N. C., and a knitting mill at Birmingham, Ala. The woodworking plants for the week are at Augusta, Ga., Hattiesburg, Miss., Shreveport, La., Dickson and Lexisburg, Tenn., Corsicana and Longview, Tex., Elwood and Portsmouth, Va. Water works are to be built at Temple, Tex., and Bedford Springs, Va. The enlargements for the week, as reported to *The Tradesman*, include flouring mills at Morgantown W. Va.; sugar mills at Iberville, La.; cotton mills at Forest City and King's Mountain N. C., and wood-working plants at Kedron, Ark., Reidsboro, N. C., and Norfolk, Va.

Machinists' Supplies and Iron.

NEW YORK, October 13, 1894.

Iron—American Pig—We quote standard brands, $12 50 to $13 for No. 1; $11 to $12 for No. 2. Southern brands, $11 50 to $12 for No. 1; $11 to $11.25 for No. 2; $10.65 to $10.75 for No. 3; $11.15 to $11.25 for No. 1 soft; $10 90 to $11 for No. 2 soft. Foundry No. 4, $10 to $10 40.

Antimony—We quote L. X., 8.35c to 8.82½c.; Cookson's, 9½c to 9¾c.; Hallett's, 7¾c. to 7½c.; and U. S. French Star, 9¼c to 9½c.

Lard Oil—Prime City we quote at 63c

Copper—The market is quiet and steady. Small second-hand lots of Lake Copper can be had at 9¾c., while all the companies ask 10c. Casting Copper is held at 9.30c. to 9.25c.

Lead—The market is weak and the volume of business is somewhat contracted. The quotations are 3½c. to 3.15c for New York delivery.

Spelter—Producers are firmer in their views and ask higher prices. Quotations are 3.45c. to 3 50c., for New York delivery. Consumers are not disposed to buy at these figures.

Tin—The market is steady. For spot 15.60c. is asked.

WANTED

"Situation and Help" Advertisements only inserted under this head. Rate 30 cents a line for each insertion. About seven words make a line. Copy should be sent to reach us not later than Saturday morning for the ensuing week's issue. Answers addressed to our care will be forwarded.

Comp. d'ftsman wants pos. BOX 7, AMER. MACH.

A mach. shop supt. of superior ability desires to make a change; cor'd'e solic'd. Box 25, AM. MACH.

Gas engine man wants situation as supt. or foreman; knows how to design and mfr. F AM. MACH.

Wanted—Mechanical, elec., civil engin'rs, draftsmen; technical work for spare time; no canvassing. Edwin Guthrie, Washington, D. C.

Situation Wanted—Tool maker and machinist, first class; capable of developing new ideas and taking charge. Box 34, AMERICAN MACHINIST.

Machine draftsman of drawing room experience wants engagement First-class skill and ability. Box 35, AMERICAN MACHINIST.

A mechanic'l engineer, tech. graduate, 3 yrs exp., desires situation. Reference to present employer. Correspondence solicited. M.M.P., AM. MACHINIST.

Wanted—A first-class man to take the foremanship of a gas engine factory. Applications must be accompanied by good references Address Gas Engine Foreman, care AMERICAN MACHINIST.

Wanted—By a 1st-class machinist and toolmaker, a position as foreman of machine shop; have held similar positions and can give best of references. Address Box 195, AMERICAN MACHINIST.

Wanted—An industrious and steady man with 10 to 15 years experience, to take charge of machine shop employing 40 men; must be well recommended. Also one good man for piping. Address Box 32. AMERICAN MACHINIST.

Wanted—A partner with $3,000, to take half interest in an established machine and repair shop; have several good patents; must be a practicable machinist, and capable of taking charge of shop. Address Partner, care AMERICAN MACHINIST.

A man wanted to instruct in machine work and pattern making in college shops. Must be experienced in wood and iron working and in forging, and come well recommended. Permanent position. Address D. C., AMERICAN MACHINIST.

Position as superintendent desired by mechanical engineer with more than 10 years' experience as such on a variety of work; record for economical production of fine work unsurpassed; first-class references Energetic, AMERICAN MACHINIST.

Position of responsibility in factory desired by a first-class reliable mechanic. Is now engaged in erecting machinery for a large engine firm. Extensive experience in water works also pumps, engines and electric machinery. Technical graduate and a young man of energy. G., AMERICAN MACHINIST.

✣ MISCELLANEOUS WANTS ✣

Advertisements will be inserted under this head at 35 cents per line, each insertion. Copy should be sent to reach us not later than Saturday morning for the ensuing week's issue. Answers addressed to our care will be forwarded.

Cheap 2d hd lathes & planers. S.M.York,Clev'd,O.

Best Steel Flue Scrapers. Kelley Co., Erie, Pa.

Calipers & Gauges. F. A. Welles, Milwaukee, Wis.

10c. let., fig. and name stamps. Payn Co., Boston.

For Sale—Second-hand drill presses, engine lathes & planers. Dietz, Schumacher & Co., Cincinnati, O.

Light and fine machinery to order; Foot Lathe Catalogue for stamp. E. O. Chase, Newark, N. J.

Wanted—To buy good second-hand tools. Box 83, AMERICAN MACHINIST.

Draftsmen's new conveniences for rapid work. Send for blue print. D. J. Kelsey, New Haven, Ct.

For Sale Cheap—A first-class foundry and machine shop. Central New York. Foundry, AM. MACHINIST.

For Sale—1 Curtis pipe machine, 2½" to 4"; 1 32" B. G. drill; 1 27 'x14" screw cutting lathe. C. M. & J. B. Hart, Clarksburg, W. Va.

Every mechanic should have a copy of "Pocket Primer of the Air Brake." Sent to any address for 50 cents. W. S. Rogers, Buffalo, N. Y.

Established in 1874.

CLEVELAND TWIST DRILL CO.

COR. LAKE & KIRTLAND STS. CLEVELAND, O.

100 & 102 Reade Street, New York.
85 Queen Victoria St., London, Eng.
5 Neue Promenade, Berlin, C., Germany.

DETROIT TWIST DRILL CO., DETROIT, MICH., U. S. A.

SOLE MANUFACTURERS OF

GRAHAM'S GROOVED SHANK TWIST DRILLS AND CHUCKS.

Endorsed by Practical Mechanics Everywhere. Send for Catalogue.

Machine and Boiler Tools,

MODERN STYLE, AT LOW PRICES.

IRON PLANERS.
19 in. x 3 ft. Wheeler.
20 " x 6 " Pratt & Whitney.
24 " x 5 " Pease
24 " x 4 " Wood & Light.
24 " x 5 " New Haven.
24 " x 6 " Wood & Light.
26 " x 7 " N. Y. S. Eng. Co
27 " x 5 " Pond
30 " x 10 " Bancroft heavy A1
36 " x 8 " Pease.
60 " x 24 " Steam Eng. Co.

SHAPERS.
16 in. Stroke Gould & Eber.
22 " " Triple Geared.

DRILLS.
3 Spindle Slate Sensitive.
3 Spindle Hendey
20 in. Swing Wheel Feed.
20 " " Prentice, stand comp
25 " " Prentice complete.
40 " " Cinti. complete
Radial Drill, 86 in. complete.

MILLING MACHINES.
Brown & Sharpe No. 6 Plain.
Brainerd Plain Table 30 x 8.
Brown & Sharpe No 1 Universal.
Lincoln Pattern, very cheap.
Garvin, No 2 and 4.

BORING MILLS.
Bement No. 2 Hor., B. & D.
Bement Cyl. Borer, 12 to 36 in.
Upright B & T. Mill, 38 in.
Brown & Sharpe Upright Turret Borer, takes in 36 x 12 in.

ENGINE LATHES.
13 in. x 6 ft. Blaisdell.
14 " x 6 " Reed Taper.
15 " x 5 " Pond.
15 " x 8 " Standard.
16 " x 6 " New Haven.
16 " x 6 " Ames.
16 " x 6 " Putnam.
20 " x 10 " Fitchburg
20 " x 12 " Prentiss Taper.
22 " x 8 " Harrington.
22 " x 10 " Putnam.
24 " x 12 " Putnam.
34 " x 8 " Putnam
35 " x 15 " Hewes & Phillips
36 " x 24 " Old Style 'cheap.'
48 " x 17 " Hewes & Phillips.

HAMMERS.
50 lb. Helve, Bradley, Up g't.
800 " Steam, Ferris & Miles.

BOILER AND BRIDGE TOOLS.
Rolls, 7, 8 and 10 ft.
Hor. Flange Punch, "H and J."
Bement Plate Planer 16 ft.
Allen Pneu. Bridge Riveter with reach of 60 in.

MISCELLANEOUS.
Imp. Brass Cabinet Turret.
Acme, 1 in. Bolt Cutter with Dies, Nut Taps, Chuck, etc.
Slotters 10 and 18 in. Stroke
Stiles No 14 Adj. Press, weight 4,200 lbs. Good as new.
Screw Machine, 2 in. hole.
Screw Machine No 2. Wire feed

J. J. McCABE,

SUCCESSOR TO
E. P. BULLARD'S
N. Y. Mach'y Warerooms.

14 Dey St.,
NEW YORK.

The "Eclipse" Drill, Tool and Graver Sharpener.

The only practical device for the re-sharpening of drills or any other tools requiring a flat surface and fine edge. It can be adjusted to any angle, and by aid of the degree or index marks the drill or tool can be set to the desired angle almost instantly.

Price, Size A $1 75, sent post paid on receipt of price.

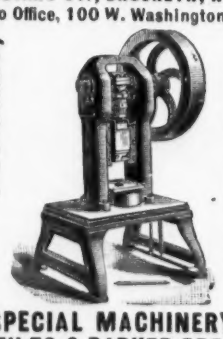
EZRA F. BOWMAN & CO.,
22 E. Chestnut St., Lancaster, Pa.

E. W. BLISS CO.

1 ADAMS ST., BROOKLYN, N. Y.

Chicago Office, 100 W. Washington Street.

PRESSES, PUNCHES, SHEARS, DIES, DROP HAMMERS,



SPECIAL MACHINERY.

The STILES & PARKER PRESS CO.

MACHINERY BARGAINS.

10 in. Post Drill.
10 " " Auto. Feed to Table. for Butt Drilling.
1, 2 and 3 Spindle Sensitive with and without Power Feed.
2, 3 and 4 Spindle Gang Drills
20 24, 26, 28, 30 and 44 in. "
60 in. Swing Post Drill.
5 ft. Arm Radial Drill.
12 in. Stroke, 16 in. x 16 in. Crank Planer.
16 in. x 4 ft. Planer.
24 " 4, 5 and 6 ft. Planer.
28 " 5 and 8 " "
30 " 5, 8 and 10 " "
60 " 20 " "
72 " 25 " "
9, 12, 16 24 and 26 in. Stroke Shaper.
10 in. Stroke Slotter.
Hyd. Rivetting Machine.
" Crane, Both Al.
Punch and Shear.
Plate Planer.

Hor. Flange Punch No. 2 H. & J. Thomson Bevelling Shear.
15 " 6 and 8 ft. Eng. Lathe.
16 " 5 6, 7 and 10 ft. Eng. "
18 " 6 7 and 8 " "
20 " 5, 6, 7, 8 and 10 ft. " "
21 " 10 and 13½ ft. " "
24 " 10 " " "
25 " 16 and 24 " " "
32 " 16 " " "
52 " 35 " " "
57 in. Double Head Driving Wheel Lathe.
88 in. x 20 ft. Engine Lathe.
Bement Car Axle Lathe.
1¼ & 2 in. Bolt Cutter Schenk's.
7 Spd'l Nut Tapper, Dunel.
2 Spd'l Profiler. No. 1 P & W.
No. 1, 2 and 2½ Screw Machs, P. & W.
No. 1 Lincoln Pattern Miller.
Hand Millers, Screw Slotters, Mill and Cutter Grinders.
No. 2 Garvin Universal Miller.

Lot of Miscellaneous Machinery, Engines, etc. Send for Latest List.

GEO. PLACE MACHINE CO.,

Offices: 145 BROADWAY and 86 LIBERTY ST.

Warehouse: 511 and 513 WEST 13th STREET,

NEW YORK.

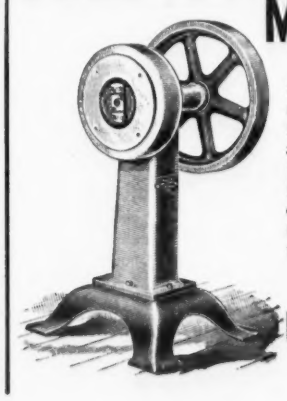
WANTED

TO BUY FOR CASH.

First-class Second-hand Engine Lathes, Planers, Drill Presses, Shapers, Milling Machines, etc., address,

The Fosdick & Plucker Machine Tool Co.,
CINCINNATI, OHIO.

COLD SWAGING MACHINE



The Dayton Swaging Machine is the best for reducing and pointing wire and tubing. If interested address the manufacturers,

EXCELSIOR NEEDLE CO.
Torrington, Conn.

AMERICAN GAS FURNACE CO.,

OIL GAS PLANTS

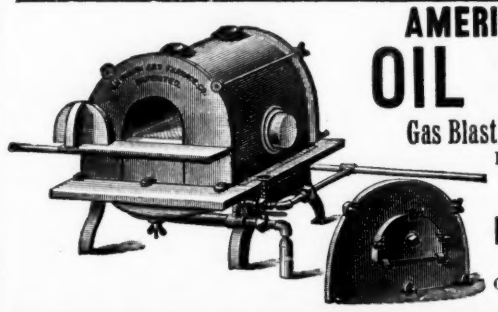
Gas Blast Furnaces & High Pressure Blowers

For the economical generation and systematic application of HEAT.

CATALOGUES ON APPLICATION.

No. 80 Nassau St., - NEW YORK.

ENGLISH AGENCY:
Chas. Churchill & Co., Ltd., 21 Cross Street, Finsbury, London, E. C., England.



SECOND-HAND MACHINE TOOLS.

ENGINE LATHES.
13 in. x 5 ft. Phenix.
15 " x 6 " Blaisdell.
15 " x 8 " Fitchburg.
16 " x 6 " Perkins.
16 " x 7 " Bement.
16 " x 8 " Perkins, Taper.
16 " x 8 " Blaisdell.
18 " x 6 " New Haven.
18 " x 6 " Bradford.
18 " x 8 " Blaisdell
18 " x 10 " "
20 " x 6 " Bullard.
20 " x 12 " Fifield.
30 " x 12 " Blaisdell.

PLANERS.
20 in x 20 in. x 5 ft. Ames.
22 in. x 22 in. x 4 ft. L. W. Pond.
24 " x 24 " x 6 " Whitcomb.
24 in. x 24 in x 5 ft. Putnam.
28 in. x 28 in. x 7 ft. New Haven.

SHAPERS.
10 in. Juengst Crank.
10 " Wood and Light Traverse Head.
15 in. Kelly Crank.
18 " Putnam Traverse Head.

DRILL PRESSES.
10 in. Milliken Bench.
10 " Gardam Bench Sensitive.
No. 1 Two Spindle Garvin.
" 2 Three Spindle Garvin.
" 3 Four Spindle Garvin.
" 3 Six Spindle Garvin.
24 in. Blaisdell.
7 ft. Alfred Box, Universal Radial.
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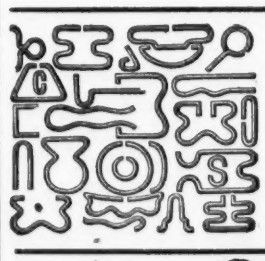

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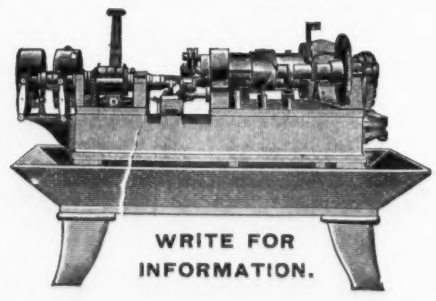

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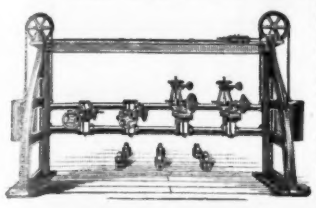

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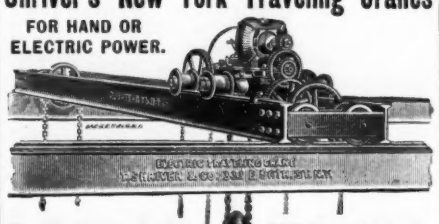

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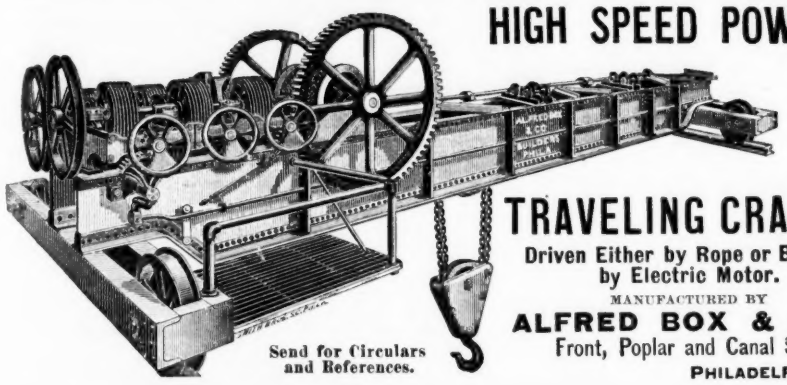

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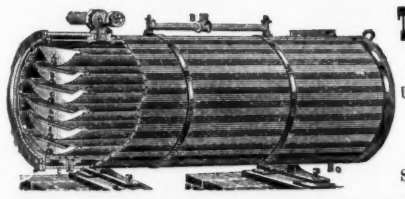

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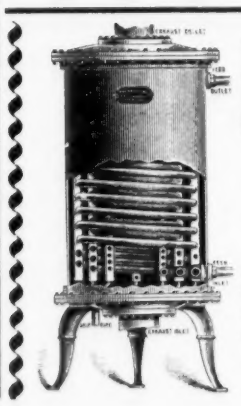

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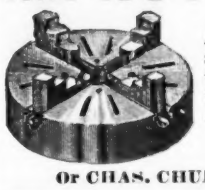

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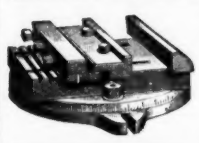

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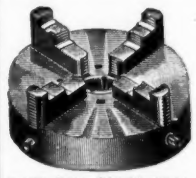

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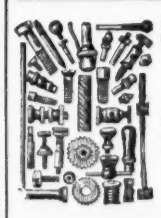

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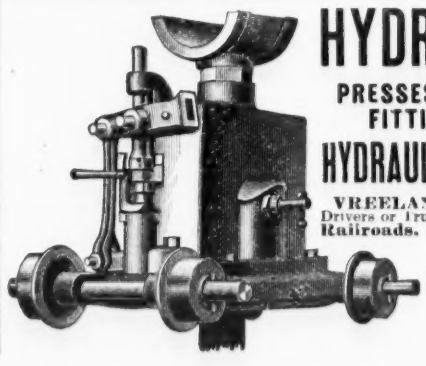

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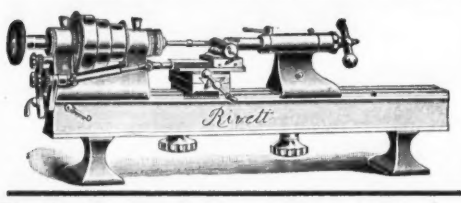

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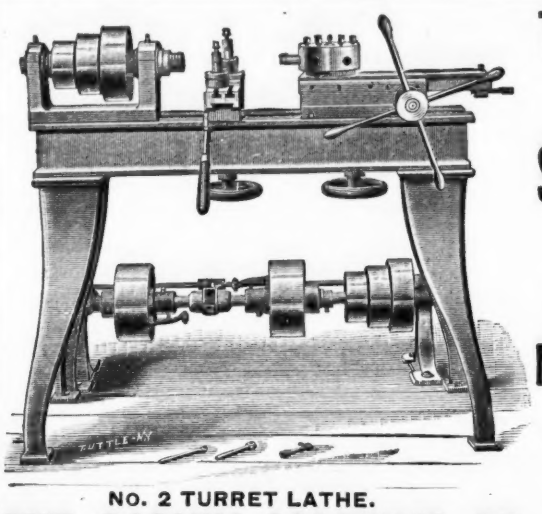

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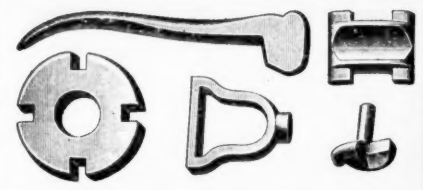

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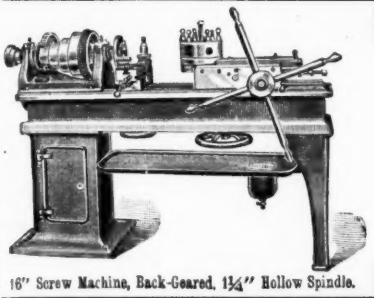

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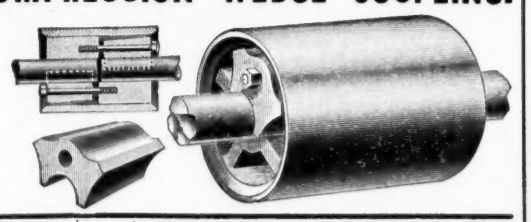

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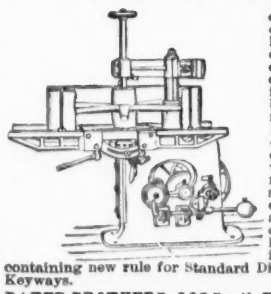

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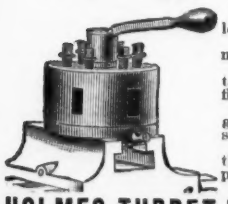

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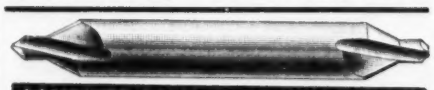

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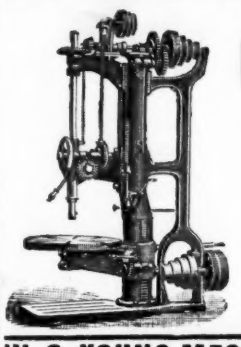

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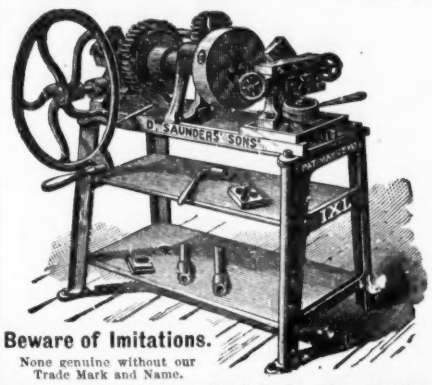

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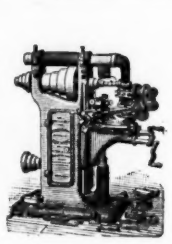

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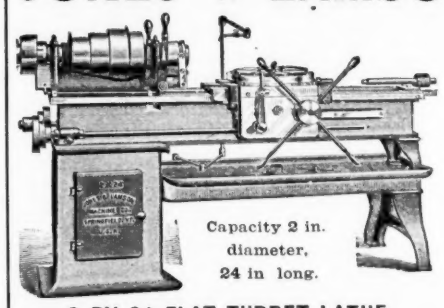

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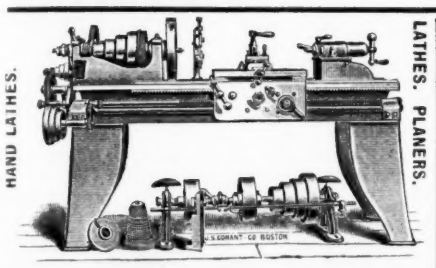

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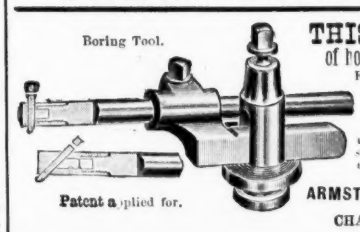

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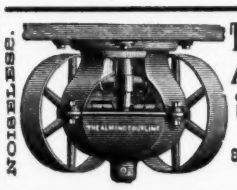

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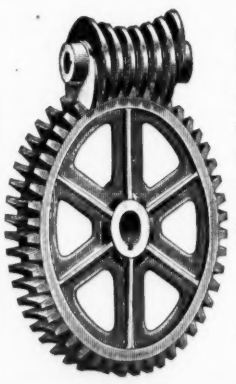

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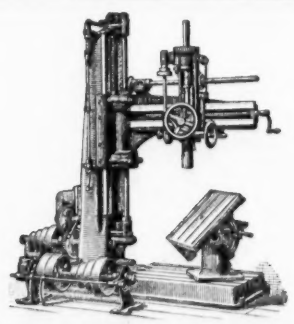

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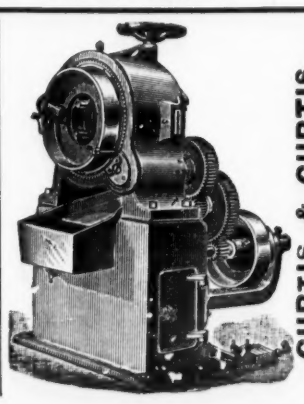

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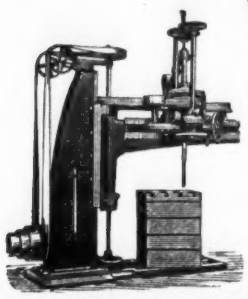

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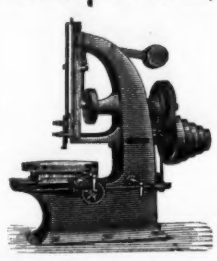

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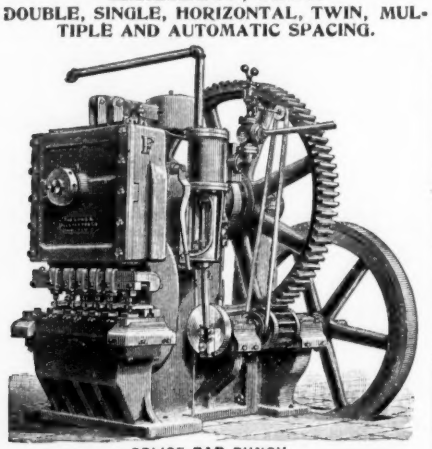

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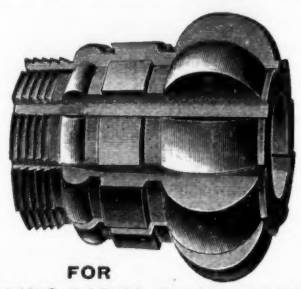

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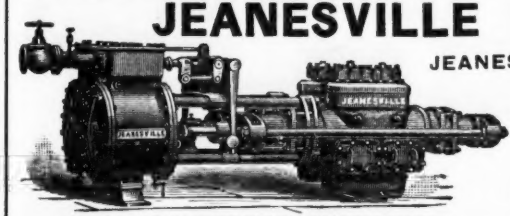

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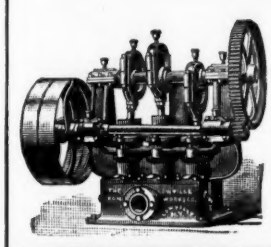

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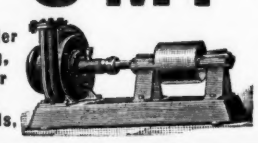

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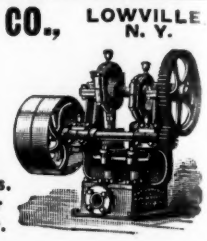

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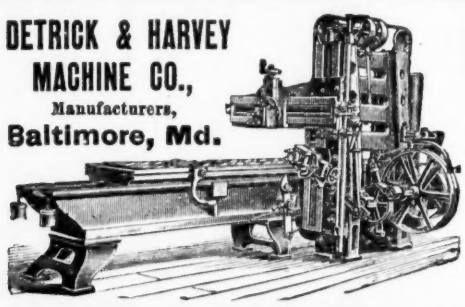

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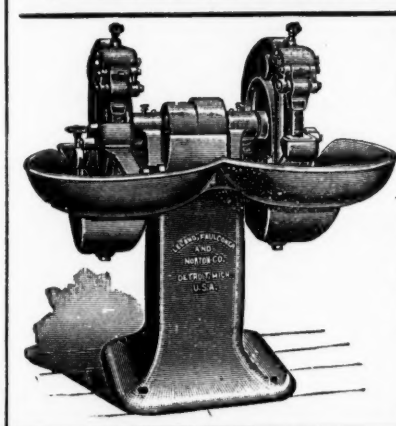

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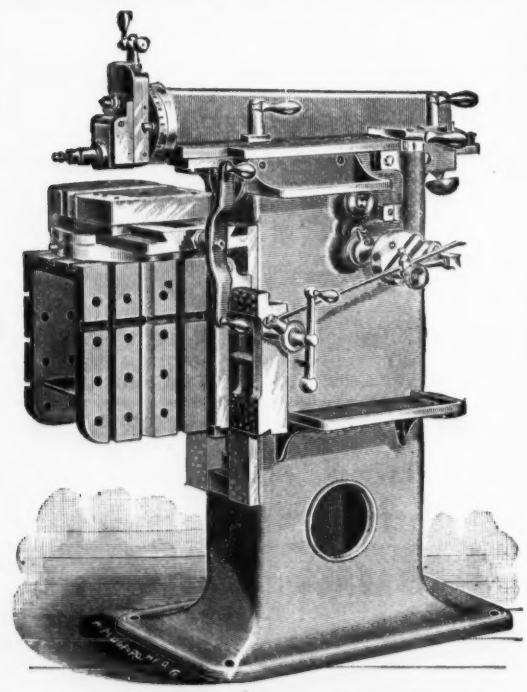

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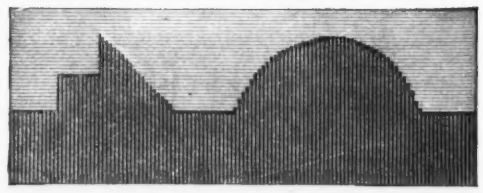

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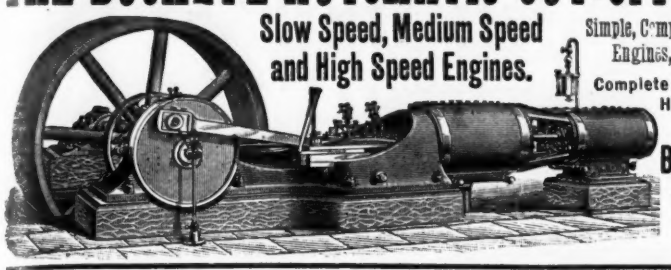

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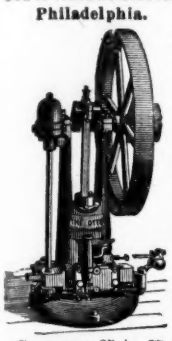

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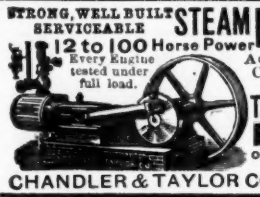

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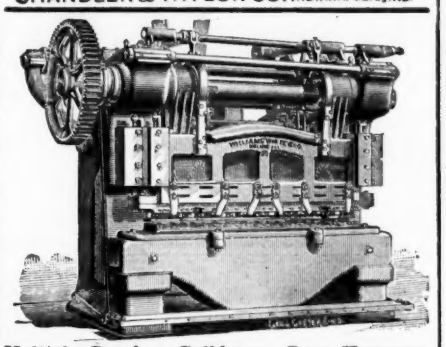

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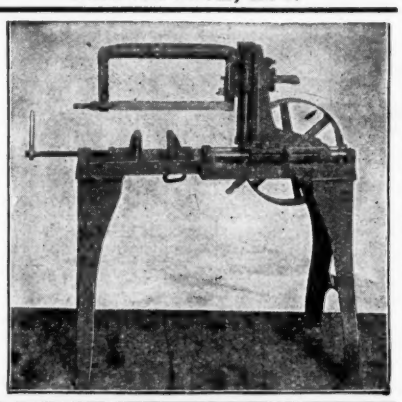

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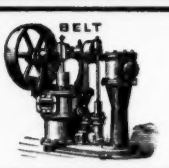

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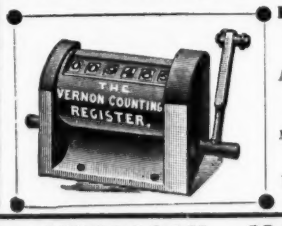

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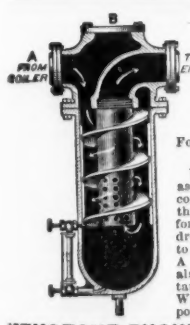

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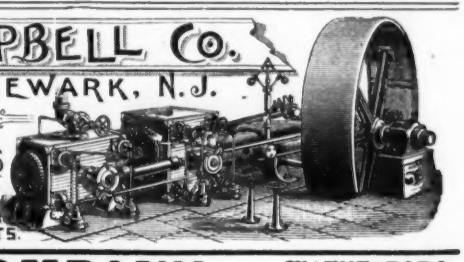

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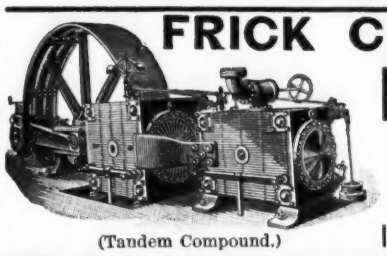

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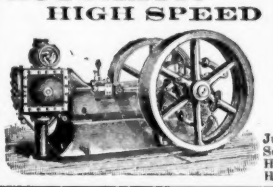

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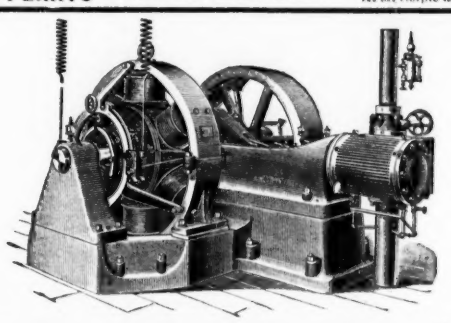

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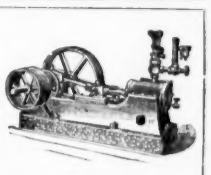

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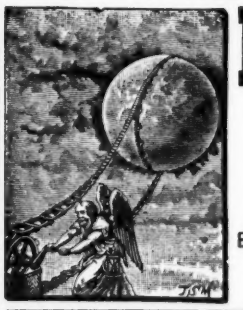

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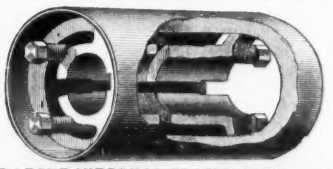

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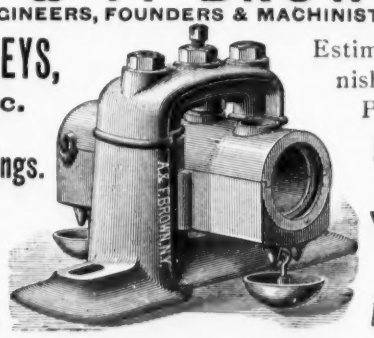

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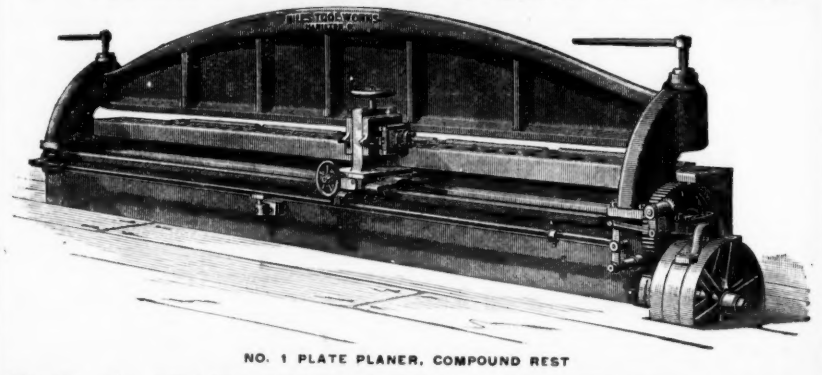

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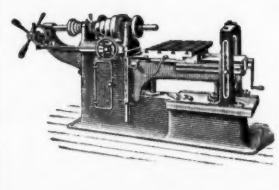

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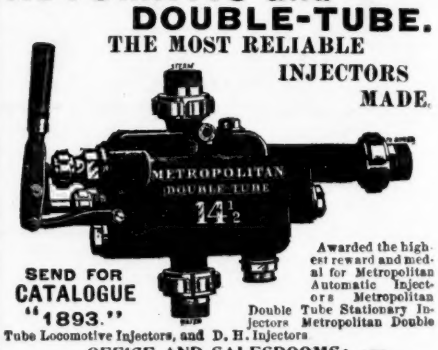

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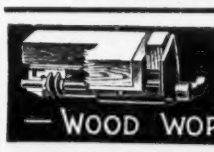

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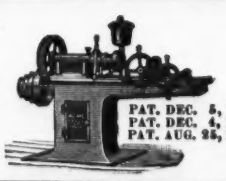

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